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The Convention Exhibit

THIS is the season of the convention and its accompanying mechanical exhibition.

The convention brings together a more or less numerous party of buyers of engineering apparatus and supplies, and affords an opportunity for the makers and distributors of these goods to create for them, in the minds of possible customers, a favorable interest.

The extent to which this is done is the measure of the real success and effectiveness of the occasion.

The engineer, whose daily routine is usually confined to a rather limited circle, finds in his attendance at a convention opportunities which, properly availed of, may be of real value to him. He has a chance to meet other members of his craft, selected men, sufficiently successful and outstanding to have been selected by their fellows as their representatives; to increase his acquaintanceship among men having the same problems and interests as his own, to learn what is going on in other plants, the results that are obtained and how they compare with his own; to increase his knowledge of men and of the world and of power-plant engineering.

And if he goes into it with the proper spirit of interest and inquiry, the mechanical exhibit, properly constituted and conducted, affords not the least of these opportunities. Here he should find the latest and best of the apparatus designed for his service. Here he should learn of new things which would improve the efficiency of his plant or offer a solution of some of his difficulties; should get a new conception of some of the standard things and find out why some are better for some uses and for what purpose each is best adapted. Here he may solve his difficulties with some forms of apparatus or perfect his understanding of those upon the function and operation of which he is not already clear.

But in order that he may do this, and in order that the exhibitor may derive an advantage in publicity and good will commensurate with the often considerable expense involved, the exhibitor must make an intelligent use of his opportunity.

The first essential to this is a real, live, well-informed representative; one who knows his goods and is capable of discussing them with engineers upon an engineering basis; one who is capable of attracting the notice and enlisting the interested attention of the passer down the aisle, and of imbuing him with the big idea behind his product; one who is patient under criticism or even abuse, tactful, engaging, interesting and convincing.

It is not enough to put up a line of finished product, however attractively arranged, and put it in charge of a boy who knows only enough to give out souvenirs. It is not enough to put your space in charge of one whose stock in trade is a plethora of trade talk which he could use in selling another's wares as well as your own. The engineers soon get the number of such a man, and their comments regarding the replies that they get from him to their engineering inquiries would dull his self-appreciation could he overhear them.

On the other hand, it is too often true that visitors come to the exhibitions in a keener quest for souvenirs than for information. The most good will come of these affairs when the delegates approach them in a spirit of inquiry and search for knowledge, and the exhibitors are awaiting them with the information right on tap.

F. R. Low

Battle Creek Steam Plant of Consumers Power Company

By W. W. TEFFT
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PLANT capacity of 27,500 kw.; modern boiler installation with large boilers to operate at high capacity; combined evaporative efficiency of boilers, water backs, superheaters and economizers of 85 per cent on test and average under load for January of 80 per cent; distinct stoker and clinker crusher drives; forced and induced draft; cast-iron, baffled economizers; high pressure two-effect evaporator system for make-up water; special coal and ash-handling with inclosed rooms to segregate dust and dirt, and complete heating and ventilating system which prevents ceiling condensation and provides preheated air for the stoker fans.

IN A previous article[1] some indication was given of the system of the Consumers Power Co., of Jackson, Mich., with its twenty hydroelectric and twelve steam plants. The generating capacities of water and steam plants are about equally divided, so that instead of using the steam plants merely as relays, it has been necessary to operate almost continuously at high load factor some of the larger and more efficient steam stations. This has been particularly true of the Elm Street Station at Battle Creek, which has been remodeled extensively within the last year, particularly in the boiler room, so that it is now the largest steam plant on the system.

The general design of the boiler plant was worked out on the basis of minimum cost per kilowatt-hour at the switchboard. In arriving at this cost, both fixed and operating charges were considered, so that too great a price was not paid for high efficiency. This method was used in determining the advisability of installing the various types of apparatus, such as economizers, forced- and induced-draft fans, type of auxiliary drive, to make sure in the first instance whether the economies effected for this apparatus were justified and, further, to determine to what degree these economies should be carried out. For instance, in the case of economizers, as in other apparatus, careful investigation was made to determine what economizer area would give the minimum total of fixed and operating costs at the switchboard.

In this way an entirely modern boiler house designed to operate continuously at capacities of 300 per cent of rating or above on high-grade West Virginia coal was obtained. Underfeed stokers with separate clinker-grinder drive and other special features are installed. Superheaters raise the temperature of the steam to 600 deg. F. Baffled cast-iron economizers insure low flue temperatures. A two-effect evaporator system supplies pure makeup water. Auxiliary drive is largely electrical. No house turbine is provided to maintain a heat balance, but instead, indirect radiation takes what little exhaust steam may be available to preheat air for the stokers, incidentally keeping the feed-water temperature low to give a greater range for the economizers. This same system heats and ventilates the turbine room and carries away the heat radiated from the turbines to be utilized in the boiler furnaces. Incidentally, it also prevents ceiling condensation. There are other features such as special coal and ash handling. Within the boiler room these operations are carried on in inclosed rooms. A wall-type motor-driven fan

[1] *Power*, April 7, 1922.

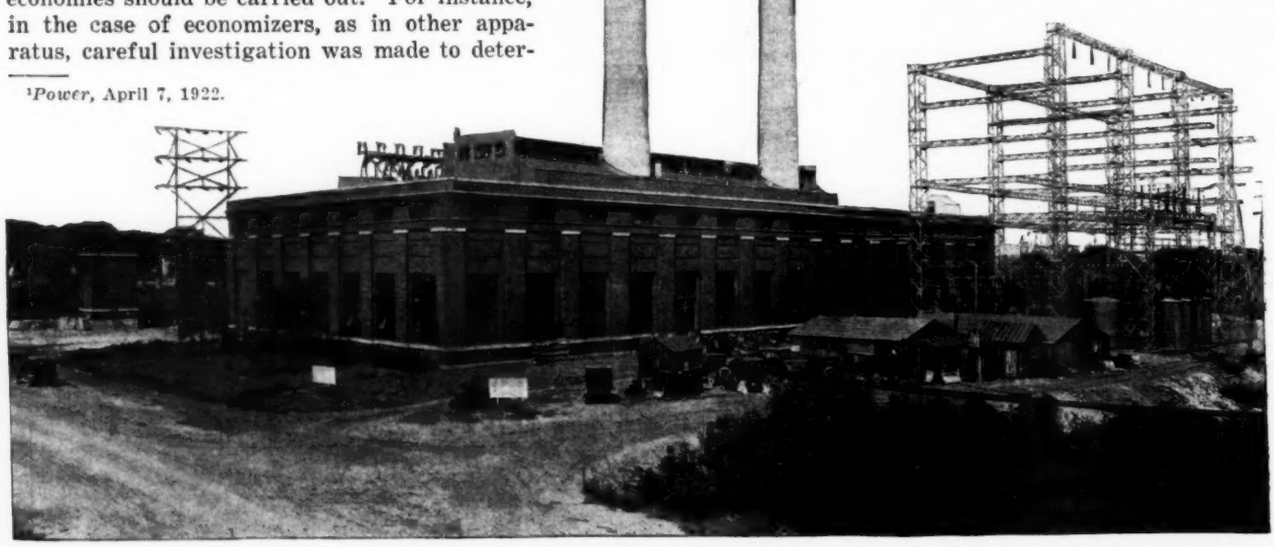


FIG. 1. GENERATING STATION OF 27,500-KW. CAPACITY OPERATES ON A BASE LOAD AT HIGH EFFICIENCY.

ventilates the ashroom. A combination duplicate feed line and loop system provides two distinct paths for delivering feed water to any boiler and also allows filling idle boilers with cold water without interfering with the supply of hot water to any of the other boilers.

A series of tests recently conducted on this equipment shows that the combined evaporative efficiency of the boilers proper, water-backs, superheaters and economizers is 85 per cent at 236 per cent of rating. The plant was operated under ordinary load conditions at a gross evaporative efficiency of over 80 per cent for the month of January, 1922. This means that over 80 per cent of the total heat in the coal as received was put into the steam pipes. All station auxiliaries, boiler-room condenser apparatus, coal handling, and every other station use require from 3 to 4½ per cent of the total power generated. This percentage depends on the load factor and ordinarily averages a little under 4 per cent. The usual method of operation is to drive all auxiliaries electrically except one radojet air ejector on each condenser requiring a maximum of 1,500 lb. of steam per hour.

In the tests referred to, at 150 per cent of rating, practically all the combustible was burned out of the ashes before passing the clinker rolls, and the flue-gas temperature at the economizer outlet was held down to about 250 deg. F. The carbon dioxide was maintained between 12 and 15 per cent.

With the ashpit and stack losses held to so low a figure and most of the auxiliaries electrically driven, the plant has operated for weeks at a time on 19,500 B.t.u. per kilowatt-hour of net plant output, equivalent to less than 1½ lb. of West Virginia coal, and at a slightly lower figure in the colder months, when the somewhat limited supply of condenser cooling water is more effective. The plant steam consumption per kilowatt-hour averages about 13 lb. and this with two 10,000-kw. units and one 7,500-kw. unit operating under a steam pressure of 225 gage and a steam temperature of 600 deg. F.

Reference to the drawings and the interior photographs shows that the boilers are arranged in a single row, each containing 8,697 sq.ft. of steam-making surface. They are of the cross-drum type with underfeed stokers set under the low end. The setting is of the Alert type commonly used with this boiler, the rear of the setting being carried in to meet the bridge wall. The superheaters are above the tubes between the first and the second pass, and the economizers are above the boiler. Above the lower tubes insulating brick reduces the radiation from the side walls, and for the same purpose the rear wall of the setting is lined all the way up with Nonpareil insulating brick. In addition, 13½ in. of firebrick protects the first pass, which, with the insulating brick and the outside face brick, gives a wall 22 in. thick. Back of the bridge wall the setting is steel-incased, with a lining of insulating brick protected by first-grade firebrick. Owing to operation at high rating and high furnace temperatures resulting, the furnace walls are not insulated. Instead they are provided with perforated brickwork having forced-draft ventilation. The bridge wall is protected by water-backs placed over the clinker grinders. Condensate from the condensers of the main generating units is circulated through these water-backs and discharged to the feed-water heater.

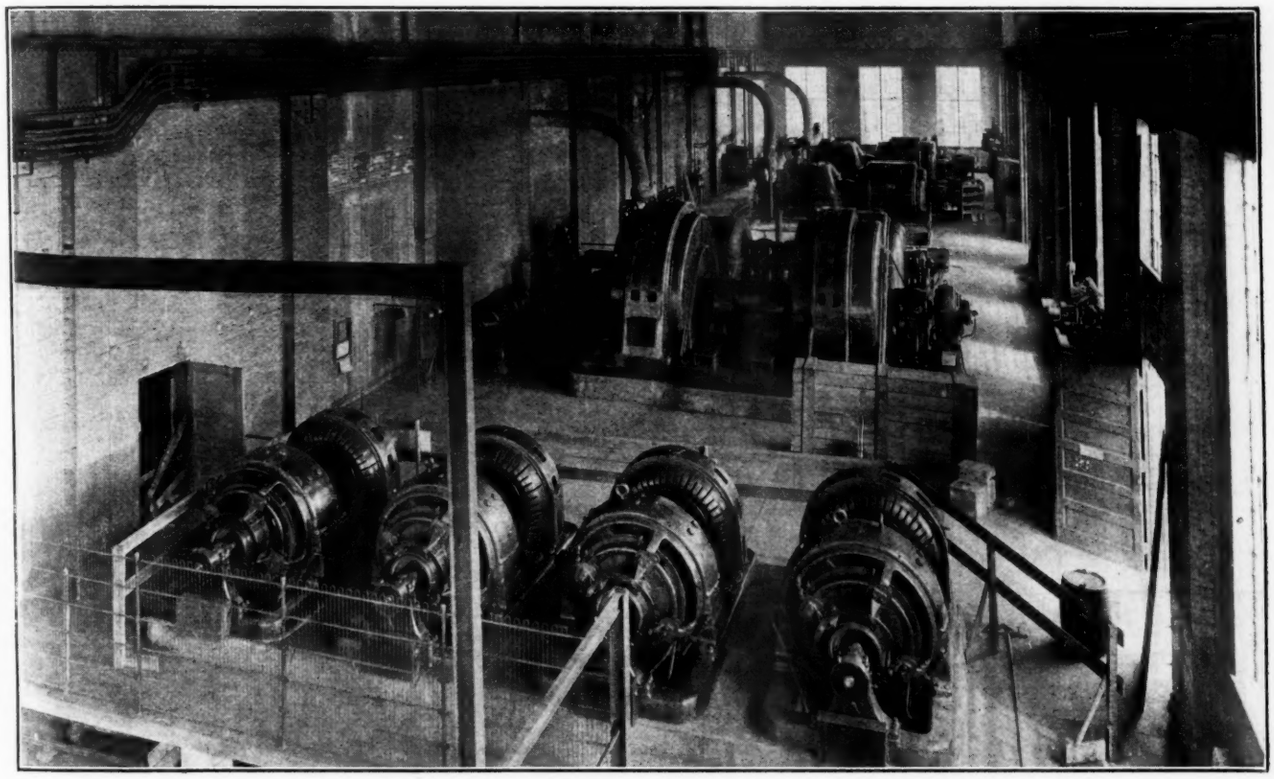


FIG. 2. THE TURBINE ROOM CONTAINS GENERATING UNITS, FREQUENCY CHANGER AND MOTOR-GENERATOR SETS

Steam is generated at 225 lb. and superheated to

give a final steam temperature at the turbine of 600 deg. F. To supply the high ratings at which it is desired to operate, each boiler is equipped with a 9-retort underfeed stoker, extra long, equipped with double-roll clinker crushers. The maximum coal-burning capacity of each retort is 1,400 lb., or 12,600 lb. per hour per stoker, this amount of coal being sufficient to evaporate over 100,000 lb. of water per hour and at a steam rate of 13 lb. per kilowatt-hour, to generate approximately 7,700 kilowatts.

Each stoker has a flexible individual drive consisting of a variable-speed direct-current motor connected by silent chain to the stoker shaft. The motors are totally inclosed, and a machine that would ordinarily be rated at 20 hp. is installed to carry a 7½-hp. load. A speed variation from 1,600 to 400 r.p.m. is obtained by means of a hand-operated street-car type drum controller. Small alternating-current motors with mechanical speed-varying devices are provided to drive the clinker rolls independently of the stoker. Should the rate of driving not correspond exactly with the quantity of ash produced, an automatic feature has been introduced to protect the rolls. Should the grinding be too fast and incandescent ash get down into the rolls, an automatic sprinkler system in the ashpit will quench the fire and the steam produced will tend to cool the rolls.

Forced draft is supplied by individual fans, each having a capacity of 45,000 cu.ft. per min. against a pressure of 6 in. and driven by variable-speed slip-ring induction motors. A speed reduction of 50 per cent, from 1,160 to 580 r.p.m., may be effected by means of hand-operated drum controllers, so that dependence is again placed on the human operator to control the air as well as the coal. Each of the fans is connected to a common duct running the full length of the boiler house, with sectionalizing dampers between each boiler and short direct ducts to the stoker windboxes. The location of the fans is in the basement in a room sealed off from the rest of the plant and made practically airtight. Entrance to the room, or exit, is effected through revolving doors.

Air for the fans may be drawn from two sources, the turbine room and from outdoors near the top of the building. A counterweighted damper in the branch, balanced against the slight vacuum in the turbine room, automatically controls the amount of outside air utilized. As operation at full capacity would renew the air in the turbine room every six minutes, in the colder months the temperature of the room would not be comfortable, so the outdoor air is tempered by means of the indirect heating coils previously mentioned. Utilizing available exhaust steam in this way preheats the furnace air, keeps the turbine room comfortable and is a factor of some importance in maintaining the heat balance of the plant. With the counterweight on the damper properly set, the amount of outside air can be regulated as desired. In the summer all the air is pulled from the turbine room, admission to the latter being through the windows or the usual channels of ventilation. With rapid circulation of the warm, dry air toward the entrance of the duct, moisture cannot be precipitated from the ceiling.

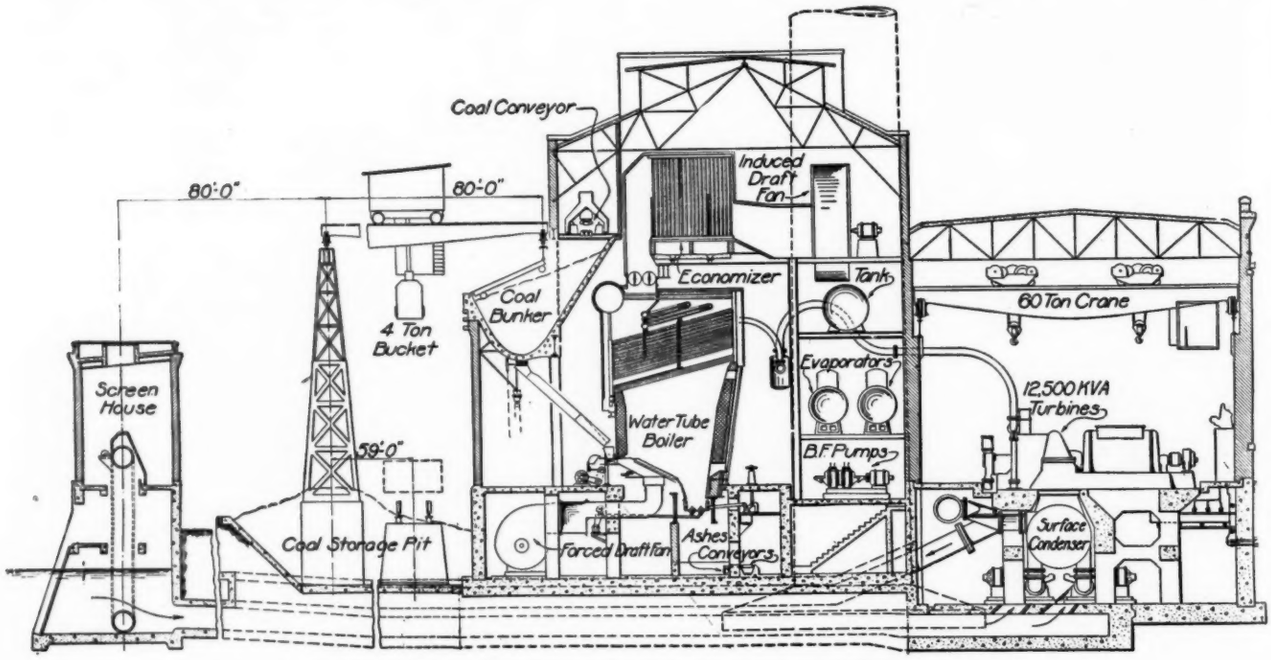


FIG. 3. TRANSVERSE SECTION OF TURBINE AND BOILER ROOM

On the other side of the basement is the ashroom, which is also sealed off to prevent dust or smoke reaching other parts of the plant. To insure good air to breathe and comfortable working conditions for the men handling the ashes, a wall-type fan is installed. The air is drawn from the turbine-room basement and discharged outdoors at the opposite end of the ash tunnel. The ash conveyor drive is also located in a closed room where it is protected from dust and ash, and this also applies to the driving mechanism of the coal conveyors.

Each stoker has an ash hopper of reinforced concrete built into the building foundation and protected by a lining of firebrick. The ash-removal system consists of a manganese steel drag chain running in a concrete

trough with a manganese steel bottom. This has been found to be entirely satisfactory and requires only one man for its operation. The drag chain runs horizontally and delivers ashes to an automatic skip hoist which discharges into an overhead bin outside of the plant. From this bin the ashes are removed by motor trucks. From the boiler ashpits the ashes are raked out into the drag conveyor which operates in a trench in the floor. A grid cover guards against the introduction of large clinker which might jam the conveyor.

At the time the boiler room was designed, coal was costing $8.50 to $10 a ton in the bunkers and the present price averages about $6.50. With a load factor averaging about 60 per cent the year around, it was realized that the installation of economizers would be a good investment. Notwithstanding a boiler pressure of 225 lb., cast-iron economizers were selected and two units in parallel were placed over each boiler. A double unit containing 6,550 sq.ft. of surface serves each boiler individually. The economizers are vertically baffled to give three passes for the gases, and the result is an exit temperature of 250 to 350 deg. F., depending on the load carried by the boiler.

Between each economizer and the smoke flue a steel-plate induced-draft fan of exceptional capacity has been installed. They have sufficient capacity to operate the boilers up to 450 per cent of rating. Two of the fans are driven by 100-hp. and four by 75-hp. slip-ring induction motors, with provision for 50 per cent speed reduction through remote control from the boiler-room operating panels. Silent chain drives connect fans and motors.

Mechanical soot blowers serve each boiler, and the same method is employed for removing soot from the economizers. The soot is blown into a pocket under the economizer, and from here a pneumatic conveying system discharges it to the ashpit of the boiler. In these conveyors air was preferred to steam to avoid the heavy maintenance probable with the latter due to the formation of sulphurous acid in the economizer refuse.

Two reinforced-concrete stacks serve three boilers each. They extend 250 ft. high above the grate.

Coal for the plant comes in over a siding having a capacity of ten ordinary railway cars per switch. The track is built on a grade, and at the center is a reinforced-concrete and steel pocket to receive the coal. The ten cars are first pushed beyond the hopper and then allowed to run back by gravity, as they are unloaded, one by one.

From the track hopper the coal is elevated on an inclined apron conveyor and is discharged onto a grizzly which separates the lump from the slack; the former, passing through a single-roll crusher and uniting again with the discharge from the grizzly, is delivered to a long inclined belt conveyor rising to the overhead bunkers above the boiler. Complete equipment, consisting of a magnetic separating unit and gratings, has been provided to remove tramp iron, rock and such refuse from the coal, which might cause obstruction of the coal chutes or damage to the stokers. Delivery to the individual 100-ton concrete bunkers serving each boiler is effected by a horizontal belt conveyor with an automatic tripper. The conveying equipment, which has a capacity to handle 60 tons of coal per hour, is driven by constant-speed squirrel-cage induction motors.

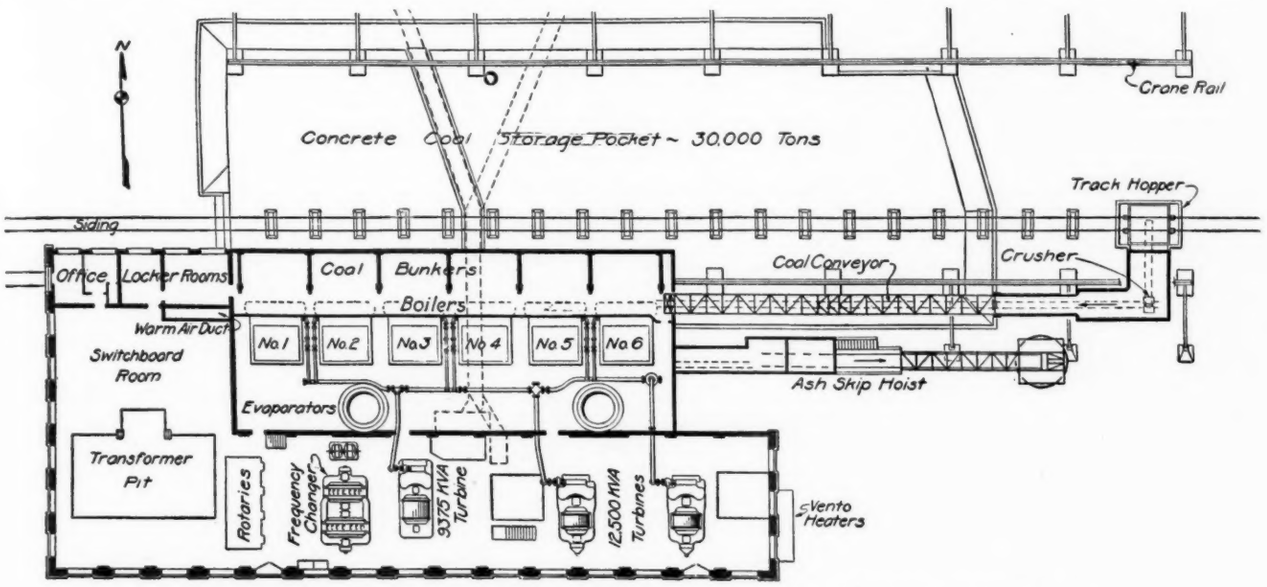


FIG. 4. GENERAL PLAN OF PLANT WITH COAL AND ASH HANDLING EQUIPMENT

An auxiliary coal-handling equipment, consisting of a bridge crane operating over the coal bunkers, railway siding and storage basin of 30,000 tons capacity, half of which tonnage may be submerged if desired, takes care of coal storage and provides an emergency means for handling coal into the plant in case the regular conveying equipment should be out of service for any reason.

From the bunkers the coal passes to the stoker hoppers through valves, chain operated from the boiler-room floor. For testing purposes there has been provided an elevated structure on wheels with a platform on which are scales and other apparatus for testing.

Ordinarily, the coal entering the bunker is weighed by a belt weightometer located near the foot of the inclined belt conveyor. The approximate rate of feeding coal to each boiler is obtained by means of counters and electric tachometers on the boiler control board. These

instruments show the speed of the stoker shaft, from which the displacement of the stoker rams may be computed and a close approximation of the coal fed secured. Ordinarily, the speed of the stoker shaft is in itself a sufficient guide for operating purposes. Over a definite period the results from the belt weightometer and the aggregate of the individual stoker records may be checked one against the other.

Each boiler has a control board on which are mounted the drum controllers for the stoker, forced- and induced-draft fans. A venturi meter records the feed water entering the boiler and a steam-flow meter gives the quantity of steam, flue-gas temperatures and the air flow all on one chart, with indication of the furnace draft as well. In addition each boiler panel has recording thermometers on the steam temperature, economizer gas outlet and on the water entering the boilers. Draft gages show the pressure of the forced draft, the furnace draft and the induced draft at the fan inlet.

At the end of the boiler room is an indicating steam gage with a 36-in. dial connected to the steam header. This gage can be seen from any point in the firing aisle and, owing to the exaggerated movement of the pointer for a small change in pressure, facilitates close regulation. At the same point is a totalizing kilowatt indicator that shows the existing load and by means of another pointer set by the switchboard operator, shows the anticipated load. Indicating thermometers are provided to show the temperature of the water leaving the water-back. A V-notch meter records all the feed water going to the boilers and may be checked with the summation of the individual venturi readings at the different control panels. Another V-notch meter measures the makeup water going to the evaporators, and the steam input to the evaporators is measured by means of a differential pressure gage.

Boiler-feed water is nearly all condensate from the main condensers, part of which passes through the water-backs in the boiler furnaces before reaching the elevated tanks. An additional tank receives the raw makeup for the evaporator system, and this supplies the evaporator feed-water heater under float control, as needed. The river is the source of the makeup water. It is taken from the condenser circulating-water outlet tunnel. From the heater a steam pump forces water into the evaporator shells. The high-pressure evaporator system has a capacity to deliver 35,000 lb. of pure water per hour, which approximately is equivalent to 10 per cent of the maximum feed-water requirements. The evaporator system has automatic feeders and automatic weirs for metering the flow, and the usual desuperheater in the live-steam supply pipe.

Pressure on the boiler-feed header is controlled by pressure-regulating valves on the pump discharges, and protection is afforded by spring-loaded pressure relief valves. In the main line to the boiler the feed is controlled automatically by regulators at each boiler, while in the auxiliary line the regulation is effected by hand.

In the turbine room the generating equipment is standard, consisting of three units, two of which are rated at 10,000 kw. at 80 per cent power factor and one at 7,500 kw. on the same basis. Three-phase 60-cycle current is generated at 5,000 volts. On the larger machines the exciters are direct-connected, and for the smaller unit excitation is supplied by either one of two exciters, one turbine-driven and the other a motor-generator set.

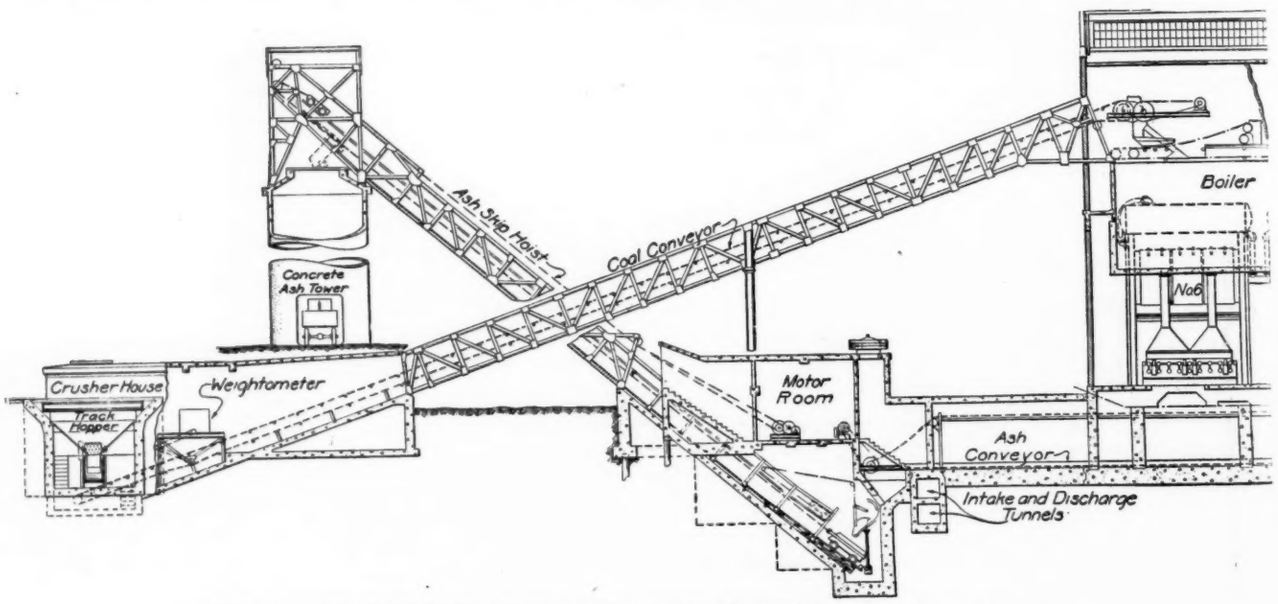


FIG. 5. ARRANGEMENT OF INCLINED BELT CONVEYOR AND SKIP ASH HOIST

Condensers are of the surface type, that for the smaller machine having an allowance of 3 sq.ft. per kilowatt of rating, or a total of 22,500 sq.ft., while the condensers for the larger machines each have 20,000 sq.ft. of surface, or 2 sq.ft. per kilowatt of generator rating.

One of the larger units has complete steam auxiliary drive so that the plant may be started in case all the transmission lines are down. The auxiliaries for the other units are driven by constant-speed induction motors. For each unit duplicate circulating pumps are installed so that one may be operated during the colder months and two in summer. Condensate pumps which are of the two-stage type are also in duplicate. Air is removed by radojets which are provided in duplicate

for the larger units and a single element for the smaller machine.

Circulating water is taken from the river in a concrete tunnel of square cross-section running under the coal-storage basin and the boiler-house foundation mat to the turbine room. The discharge is carried back to the river through a similar concrete duct. At the intake trash racks and two traveling screens remove débris that is too large to pass through the condenser tubes. Auxiliary stationary screens are also provided. Very little débris except leaves in the fall is contained in the water, so that the screens are revolved only for short periods at infrequent intervals.

In designing the electrical layout one of the principal thoughts in mind was to avoid defeating in any degree the purpose of the economizer installation. This meant keeping the feed-water temperature entering the economizer at the lowest safe temperature, which is about 120 deg. F., and also the elimination of the house-turbine so commonly used in recent installations. A duplicate bus system, which can be supplied with 440-volt three-phase alternating current, is provided. The auxiliary arrangements are made to supply these buses with power from either one or two groups of three transformers each. These auxiliary transformers can be supplied from any of the several sources of power available, there being three generating units in the station and five transmission lines connecting the station to outside sources of power. With so many alternates available, reliable power for the auxiliaries is assured.

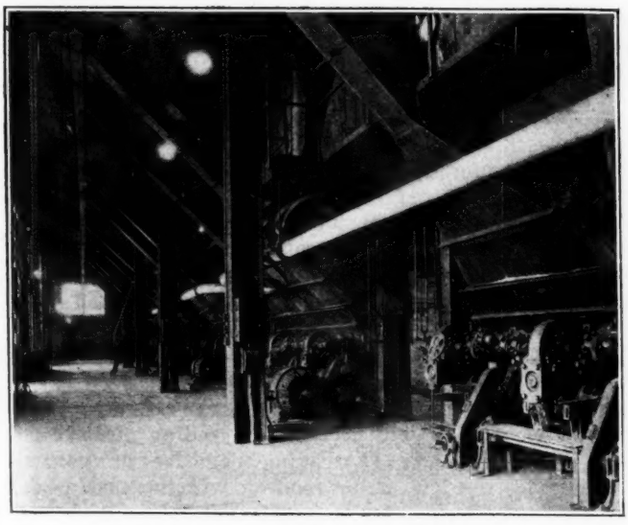


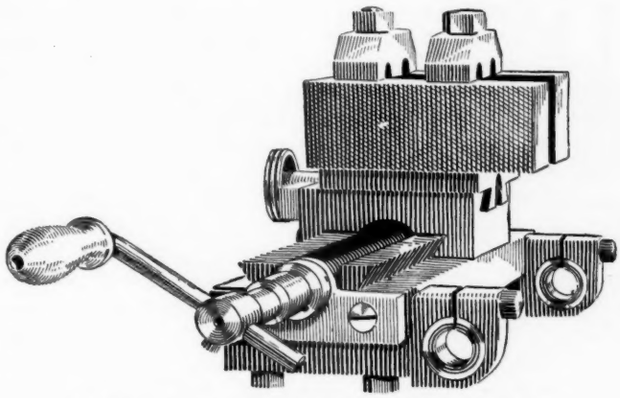
FIG. 6. VIEW OF FIRING AISLE

Direct current, which is used only for stoker drive, is obtained from two 100-kw. rotary converters. Either one of these rotaries has sufficient capacity to take care of the maximum direct-current requirements.

Among the bits of useful information in the 1921 Annual Safety Congress Proceedings is the suggestion offered by R. R. Talbot, of Fred T. Ley & Co., Inc., who told the construction section that in his company all parts of equipment that are supposed to be covered or protected by a guard are painted red. Thus when any such guard is removed the bright red underneath is apparent and everyone in the vicinity knows that a safeguard has been removed and ought to be replaced. —*National Safety News.*

Commutator Grinding Tool

Grinding of commutators has become general practice, with a decided improvement over the old method of using sandpaper. Grinding is better than turning with a lathe tool, because the work can be done under full-speed operating conditions and the cost is less. If the commutators are watched and trued up whenever



ASSEMBLY OF COMMUTATOR-GRINDING TOOL

they start to go bad, the necessity of truing in a lathe may be practically eliminated.

Hand stones in various forms have been used to a large extent, and where the commutator is not badly eccentric these will produce a true surface and give satisfactory results. However, where commutators are large and have large flat spots or eccentricity, it is advisable to support the stone in some form of stationary rest.

For the latter conditions the Martindale Electric Co., Cleveland, Ohio, has developed and placed on the market the commutator grinding tool shown in the figure. The device is made to be supported on one of the brush-holder studs, and is designed so as to permit grinding up to the commutator risers. A special adjustable support is furnished for clamping the grinder to flat brush-holder arms. The tool operates on the principle of a lathe-tool rest; and provisions are provided for lateral movement of the stone across the commutator's face as well as adjustment of the stone toward the commutator for grinding depth.

Speaking of the tremendous extent to which the United States is dependent upon the energy obtained from its material resources, C. J. Gilbert and J. E. Pogue in their recent book "America's Power Resources" make the following statement: "The actual figures are so staggering in their magnitudes that a further comparison is necessary if their full significance is to be grasped. The amount of work which an average man can perform in a day is a convenient and familiar unit. It would require the labor of three billion hard-working slaves to accomplish the work done annually in the United States by our energy resources. This estimate is a rough approximation merely, based upon an assumed power utilization of 150,000,000 hp. and the equivalence of 20 man-power for one horse-power. The use of energy materials gives to each man, woman and child in this country the equivalent of thirty servants. Our type of civilization arises from this organized use of inanimate energy."

Power-Factor Correction by Rearrangement of Motor Drives

By T. H. ARNOLD
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DURING the stress of war conditions when deliveries of equipment were slow and uncertain, a tendency developed to use any available apparatus that would keep the wheels turning. In the interest of conservation of equipment already manufactured and thus relieving the manufacturing capacity of the country for more strictly war purposes the use of obsolete equipment was entirely justified. The machines also had the additional advantage of being immediately available. In many cases motors were installed with wide variations in load from the rated capacity. With a direct-current system the most serious defects are shortened life due to overloads and loss of efficiency due to carrying underloads. An alternating-current system will have the additional disadvantage of a low power factor from underloaded motors. The disadvantages of low power factor are too thoroughly known to require discussion. The present time, while many industries are running at reduced capacity, offers an excellent opportunity for the investigation and study of these conditions with a view toward bettering them as operations are resumed on a normal basis.

FOUR methods of power factor correction.—A plant load of 15 motors having a combined capacity of 465 hp. operating at a power factor of 0.76, is taken as an example, and the problem of rearranging these motors so as to do the work with 290-hp. capacity installed is worked out in detail, obtaining a power factor of 83.9 per cent.

The power factor of a system can usually be increased by one or a combination of the following methods:

1. By rearranging the motors to operate at approximately full load.
2. By installing synchronous motors on some drives to operate at or near 100 per cent power factor.
3. By installing synchronous motors to drive mechanical load and also supply leading current to the system. The most economical combination is 70 per cent mechanical load and 70 per cent leading current. This gives a resultant combination equal to 100 per cent load on the motor.
4. By installing static or synchronous condensers at load centers to correct the power factor.

From the viewpoint of efficiency and economy the first method generally offers the greatest return with the lowest outlay and should receive first consideration. A careful check on the power consumption of each motor should be made, taking maximum, minimum and average kilowatts of load, voltage at the motor and power factor. With three-phase motors the most convenient method for obtaining the power factor if a portable power-factor meter is not available, is to take the high and low single-phase wattmeter readings while taking the total power. With a polyphase wattmeter this may be done by disconnecting one side of the meter and taking the reading and then repeating the operation with the other side. These readings have a definite relation with the power factor so long as the load is balanced, which it is in most cases with polyphase motors. A formula for obtaining the power factor from the wattmeter readings on a three-phase load is:

$$\textit{Power Factor} = \frac{W_1 + W_2}{2\sqrt{W_1^2 - W_1W_2 + W_2^2}}$$

Where W_1 equals high reading and W_2 equals low reading of the wattmeters. Plotting the values obtained by dividing the low reading by the high reading against the power factor, a curve, as shown in Fig. 1, is derived, from which the individual power factors of the motors can be read. This method will usually be sufficiently accurate for commercial testing and will probably be within the limits of voltage and load fluctuations. Below 50 per cent power factor the low reading becomes negative, the wattmeter reverses, and increases to a negative maximum as zero power factor is approached.

As an example, suppose two single-phase wattmeters are connected on a three-phase motor as shown in Fig. 2 and that one meter reads 80 kw. and the other 50 kw. Dividing the low reading by the high gives $50 \div 80 = 0.625$. Where the two meters read reasonably close together, as in this case, the low reading is positive, unless the power factor is exceedingly low. If one is reasonably sure that the motor is carrying a considerable load, then unless the ratio of the two readings becomes around 0.2, it is safe to consider the readings positive. However, if in doubt take the reading with the load on the motor and with the load removed. If the low reading meter is reading positive when the motor is driving its load, it will reverse and throw back off the scale when the load is removed. Or if only part of the load can be removed, the low meter reading will decrease if reading positive on normal motor load. If reading negative on normal motor load, the low meter reading will increase when the load is reduced. In the problem it is assumed that the readings are found to be positive and would be true unless the motor was of large capacity. Locating 0.625 on the right-hand side of Fig. 1 and following across horizontally to intersect the curve, then drop down vertically to the base, as indicated by the dotted lines, and the power factor is found to be approximately 0.93. By the formula:

$$\textit{Power Factor} = \frac{W_1 + W_2}{2\sqrt{(W_1^2 - W_1W_2 + W_2^2}} = \frac{80 + 50}{2\sqrt{(80^2 - (80 \times 50) + 50^2)}} = 0.93,$$

which checks with the value obtained from the curve, Fig. 1.

Tests made on a certain plant resulted in the tabulation given in Table I. A study of this table indicated

changes which appeared advisable and are given in Table II. After the test data in Table I were obtained, a power-factor diagram (Fig. 3) for the system was made. This diagram is very easily constructed. First draw the base line *AX* and lay off on this line to scale distance *AB* equal to the total load in kilowatts, or in this case 187.8. Next divide this line up into sections equal to the individual loads in kilowatts, such as load No. 1 equals 23.9, load No. 2 equals 35.1, etc. At point *C* erect a perpendicular and from *A* draw *AD* to intersect the perpendicular at *D*, forming the right triangle

TABLE I. ORIGINAL MOTOR ARRANGEMENT

No.	Speed R.p.m.	Rated Hp.	Ave. Hp.	Load Kw.	Max. Hp.	P.F.	Kva	Drive	Remarks
1	690	100	32	23.9	41	0.74	32.3	Belted	Constant
2	690	100	47	35.1	55	0.81	43.4	Belted	Fluctuates
3	1,120	5	7	5.2	7.5	0.87	6.0	Belted	
4	690	10	4.5	3.4	6	0.58	7.9	Belted	
5	690	30	28	20.9	32	0.83	25.2	D. C.	
6	1,120	15	18	13.4	19	0.90	14.9	D. C.	Constant
7	1,120	20	17	12.7	19	0.91	14.0	D. C.	Constant
8	690	30	8	6.0	11	0.48	12.5	Belted	
9	1,120	20	6	4.5	12	0.70	6.4	Belted	
10	690	10	5	3.7	6	0.60	6.2	Belted	
11	690	20	19	14.2	21	0.83	17.1	Belted	
12	690	30	18	13.4	22	0.75	17.9	Belted	Intermittent
13	690	15	8	6.0	20	0.65	9.2	Belted	Short peaks
14	1,120	10	15	11.2	19	0.90	12.4	Belted	
15	850	50	19	14.2	20	0.60	23.7	Belted	
		465	251.5	187.8		0.767	244.8		Totals

ACD. The angle between *AC* and *AD* is obtained by referring to a table of natural sines and cosines and selecting an angle whose cosine is equal to the power factor of the motor. The power factor of motor No. 1 in Table I is 0.74, which corresponds to the cosine of an angle of 42.25 deg. The length of *AD* will represent to scale the kilovolt-amperes taken by the motor.

Next, from *D* draw *DE'* parallel and equal to *CE*. From *E'* erect a perpendicular and draw *DF* to intersect it at *F*, thus forming the right triangle *DE'F*. Angle *FDE'* is made equal to an angle whose cosine equals the power factor of motor No. 2. Power factor

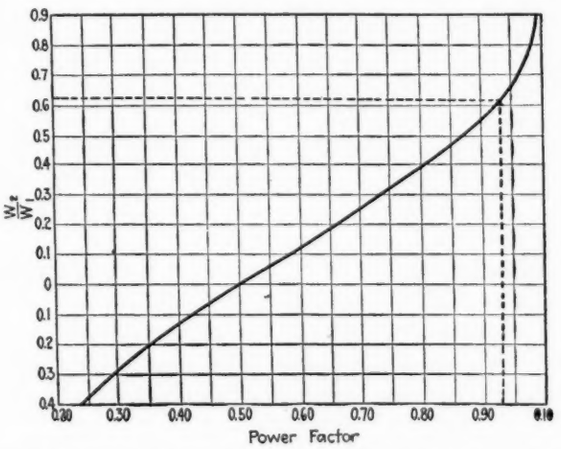


FIG. 1. CURVE FOR OBTAINING POWER FACTOR ON THREE-PHASE CIRCUIT FROM TWO WATTMETER READINGS

of motor No. 2 is 0.81 and corresponds to the cosine of an angle of 35.75 deg. This process is followed for each motor, and the result is a series of triangles as shown by the full lines. When the last motor load has been added, point *G* will be in a vertical line with *B* and by joining points *A* and *G* the line *AG* will represent to scale the kilovolt-ampere load on the system, or in Table I 244.8. Cosine of angle θ is the accumulative power factor of the system, or in this case cosine 40 deg. equals 0.767 as given in Table I. The accumulative power-factor chart for the rearrangement of motors is shown in dotted lines Fig. 3.

A method of constructing the power-factor chart, Fig. 3, that is probably a little more accurate than that given in the foregoing is one that involves calculating the reactive components of the induction-motor loads. Where the horizontal line *AB* represents the kilowatt load, the vertical line *BG* represents the total reactive component of the load. The power factor is the cosine of the angle of phase displacement between the current and voltage, where the reactive factor is the sine of this angle. The load in *Kw.* = *Kva.* × *Power Factor;* likewise, the *Reactive Component* = *Kva.* × *Reactive Factor*. For example, in load No. 1, in Table I, the kva. load is 32.3 and the power factor is 0.74. Referring to a table of natural sines and cosines, it is found that 0.74 corresponds to the cosine of an angle of 42.25 deg., and the sine of this angle is 0.67, which is the reactive factor. Then for No. 1 load the *Reactive Component* = 32.3 × 0.67 = 21.64 kilovolt-amperes. Then if distance *BD'* is laid off on the vertical *BG* equal to 21.64, a horizontal line drawn to the left from *D'* will intersect the vertical from *C* at *D*, and a line drawn from *A* to *D* will represent the kilovolt-ampere load as shown in the figure. For load No. 2, kilovolt-ampere equals 43.4 and the power factor 0.81. This power factor corresponds to the cosine of an angle 35.75 deg., the sine of which is 0.584, and this

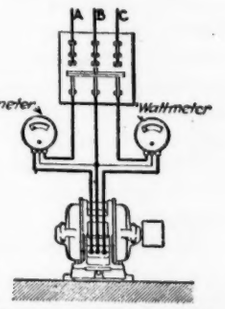


FIG. 2. TWO SINGLE-PHASE WATT-METERS CONNECTED ON THREE-PHASE CIRCUIT

TABLE II. MOTOR REARRANGEMENT RECOMMENDED

No.	Changed From No.	Speed R.p.m.	Rated Hp.	Ave. Hp.	Load Kw.	Max. Hp.	P.F.	Kva.	Drive	Remarks
1	8	690	30	32	23.9	41	0.84	28.5	Belted	Constant
2	15	850	50	47	35.1	55	0.87	40.3	Belted	Fluctuates
3	4	690	10	7	5.2	7.5	0.75	6.9	Belted	
4	3	1,120	5	4.5	3.4	6	0.85	4.0	Belted	
5	5	690	30	28	20.9	32	0.83	25.2	D. C.	
6	9	1,120	20	18	13.4	19	0.91	14.7	D. C.	Constant
7	7	1,120	20	17	12.7	19	0.91	14.0	D. C.	Constant
8	10	690	10	8	6.0	11	0.74	8.1	Belted	
9	14	1,120	10	6	4.5	12	0.86	5.2	Belted	
10	New	1,120	5	5	3.7	6	0.86	4.3	Belted	
11	11	690	20	19	14.2	21	0.83	17.1	Belted	
12	12	690	30	18	13.4	22	0.75	17.9	Belted	Intermittent
13	6	1,120	15	8	6.0	20	0.86	7.0	D. C.	Short peaks
14	13	690	15	15	11.2	19	0.82	13.7	Belted	
15	New	690	20	19	14.2	20	0.83	17.1	Belted	
			290	251.5	187.8		0.839	223.6		Totals.

Motors not required in rearrangement No. 1, and No. 2 (2-690 r.p.m. 100 hp.)
New motors required No. 10 and No. 15 (1-1,120 r.p.m. 5 hp. and 1-690 r.p.m. 20 hp.)

is the reactive factor of the load. Then the *Reactive Component* = 43.4 × 0.584 = 25.34 kilovolt-amperes. Laying off *D'F'* equal to 25.34 on *BG* gives point *F'*, from which a horizontal can be projected which will intersect the vertical from *E* at *F*. This will give the three vertices of the triangle *DE'F*. Then the triangle may be completed just as in the method previously described. This process can be followed until the whole chart is completed.

Some tests will be necessary to determine the power-factor characteristics of the different types of motor, in case this is not already known, in order to estimate the results to be derived from the rearrangement in advance, or this information may be obtained from the manufacturers.

The results for each motor should be combined to form the chart for the individual feeders and the feeder charts combined to form a system chart. The motors on the feeder shown in Fig. 3 and Table I were regarded as well within good loading practice with the exception of two of the larger motors, which were known to be underloaded and a small motor which gave evidence of heating and was suspected of being overloaded. Thus the results of the tests were in the nature of a surprise to the operating management. The net results obtained by the purchase of one 5-hp. and one 20-hp. motor and the rearrangement of the rest of the motors gave results very closely approximating those anticipated. The total rated horsepower of the motors was reduced from 465 to 290. The kilovolt-amperes reduced from 244.8 to 223.6 and the power factor was increased from 0.767 to 0.839. The resale value of the two 100-hp. motors released should pay for the entire change.

In case an investigation of the motors does not indicate that improvement can be made by their rearrangement, an investigation should be made to determine how far the installation of synchronous motors on some of the drives is economically justified. The synchronous motor has a comparatively low starting torque. Recent developments in starting windings and magnetic clutches have largely eliminated this defect. From the viewpoint of installation cost, this type of motor should be confined to the medium-sized and large drives. From the viewpoint of satisfactory operation they should be located where operators of some skill are available for starting and field adjustment. In considering possible drives of this nature, the effect on the system as a whole should be considered in addition to the individual motor. The substitution of a synchronous motor may increase the power consumption a small percentage on the individual drive, but this increase may result in a saving on the system as a whole which will greatly overbalance the loss.

Mining loads, which can usually be investigated as possible for synchronous-motor drives, are air compressors, direct-current generators, ventilating fans, smelter blowers, large pumps, and with the aid of magnetic clutches mills of the ball or tube type; in general, any of the large applications that have a steady all-day load. In case of excessive starting duty the magnetic clutch or a special starting winding will often solve the difficulty. The standard synchronous motor is built to carry its load between 0.80 and unity power factor. This in many cases will give sufficient power-factor correction. In case it is necessary to carry the correction considerably beyond unity on the motor and operate it with a leading power factor, it may be necessary to specify heavier field coils. As these motors represent a considerable financial outlay, it is seldom feasible to change an existing installation. Where new motors in the larger sizes are being considered, the synchronous type should receive careful study in this connection.

In driving air compressors, in particular, where the load fluctuates, the flywheel effect should be ample to avoid critical stages where pumping of the current will develop between the motor and other apparatus on the line. In one case in mind severe sparking would start on the direct-current side of some rotary converters in the same substation. This was accompanied by a rapid fluctuation of the alternating current from zero to the maximum shown on the alternating-current ammeter. In some cases this continued until the rotary switch kicked out, and was traced to an interchange of current between the rotaries and synchronous motor of the air compressor. This appeared only at certain load conditions on the compressor and disappeared as the load was either increased or decreased, and it seemed to be independent of the load on the rotary. Recalculation of the compressor unit revealed that the critical condition did exist and an increase in flywheel weight was the only remedy.

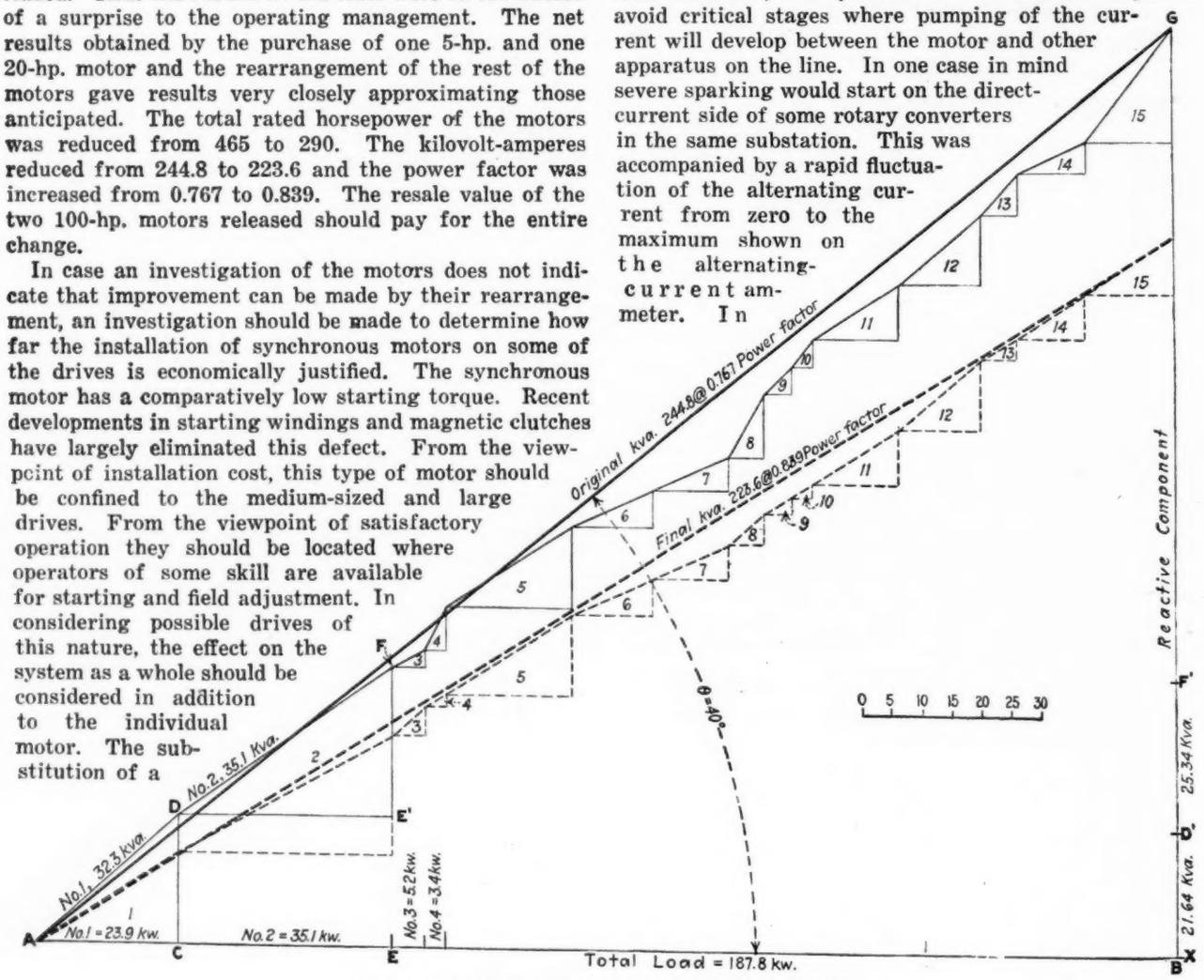


FIG 3. POWER FACTOR DIAGRAM FOR 15 MOTORS

The synchronous condenser is a variation of the synchronous motor which is built to carry its rated kilovolt-amperes in a stable manner at practically zero power factor leading. It can never quite reach that point on account of the internal losses of the machine, but for all practical purposes it may be considered as zero. This machine may also carry some mechanical

load. The static condenser will likewise have practically a zero leading power factor. This is composed of a number of condenser units mounted in a frame and connected in series or series parallel. In order not to consume too much material and make the apparatus too expensive, these units are built to operate usually at 2,200 volts, and where they are connected to lower voltage systems, transformers or auto-transformers are used to step the voltage up. The cost is favorable in the smaller capacities up to 300 to 500 kva., beyond which point it will ordinarily be more economical to install rotary apparatus. The static type has the advantage of requiring little operating attention beyond an occasional inspection and will show somewhat lower power losses.

In considering the corrective effect it is not economical to attempt to correct the system power factor too high. Each system should be considered on its merits and the economical operating point determined; 80 per cent to 85 per cent is usually satisfactory, and it is seldom advisable to go beyond 90 per cent. The amount of correction should be an economic balance of the line, transformer and generator losses against the interest on the investment, condenser losses and the cost of operation. Obviously, conditions which will justify the installation in one case may not in another.

Two-Stroke-Cycle Diesel Cylinder Heads*

BY L. J. LE MESURIER

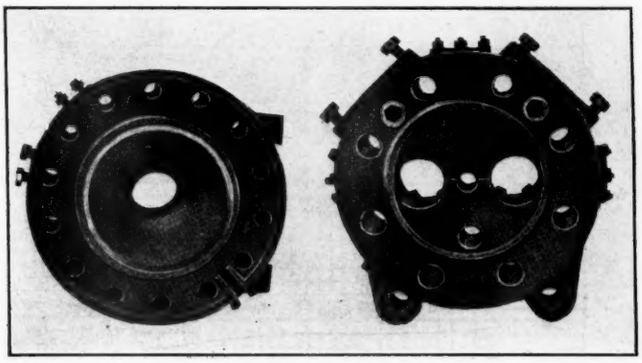
There are at least 1,500 two-stroke-cycle Diesel cylinders in service with the single fuel-valve opening, and of these over 400 cylinders are above 20 in. in diameter. The mean indicated pressure is generally about 100 lb. per sq.in., and the power per cylinder in several cases is over 750 indicated horsepower. If there was some defect inherent in the two-stroke-cycle design, as, for instance, if heat stresses had not been taken properly into account, there would surely be a considerable percentage of failures due to cracked heads. The fact that this has not been the case is the clearest proof that the design of the symmetrical head is sound both in theory and practice.

Another point of advantage associated with this design of two-stroke-cycle cover is that owing to the slightly concave shape of the under side, taken in conjunction with a similarly concave-shaped piston head, a well-shaped combustion space is obtained, contributing to efficient burning of the fuel.

It is sometimes assumed that the mechanical efficiency or ratio of brake horsepower to indicated horsepower of a two-stroke-cycle engine must necessarily be lower than for a four-stroke-cycle engine owing to the power required for driving the scavenging pump. It must be remembered, however, that the moving parts of a four-stroke-cycle engine are for the same power considerably heavier and both the number and sizes of bearing surfaces greater than in a two-stroke-cycle engine, thus adding to frictional losses. Furthermore, pumping losses are incurred during the inlet and exhaust stroke of the four-stroke-cycle engine, owing to the frictional and eddy resistance of the air and exhaust products passing through the valves and pipes.

With regard to the limits of power of the Diesel engine there appears to be in some directions a tendency to favor double-acting engines for large marine units, presumably on the grounds that by this means larger powers can be developed. The double-acting engine has, however, its own problems, which remain to be solved by experimental means and by observation of actual results in service, whereas there is a vast amount of experience behind the single-acting engine.

The combustion space in a Diesel engine is bounded by the cylinder cover, piston and the upper portion of the cylinder liner, or in the case of an opposed piston engine by two pistons and a portion of the cylinder liner. The temperature of combustion is about 2,500 deg. F. to 3,000 deg. F. and falls during the working stroke to about 1,200 deg. F., these figures varying, of course, according to the relative weight of the fuel and air charge. The heat flow through the material of the cylinder liners, piston and cylinder head, gives rise to stresses in the material which become more serious as the size of cylinder increases. The cylinder cover has always proved the greatest difficulty, as the casting is usually the most complicated and at the same time is subjected to the maximum temperature and pressure conditions. The cylinder cover and piston are together responsible for over 80 per cent of total heat flow to the cooling medium, the remainder of the heat being carried away through the cylinder liner. Although the water- or oil-cooled piston of a large engine carries away at least as much heat as the cover, the casting is much smaller than the cover casting, and freedom for expansion is more easily obtained. The liner of the normal type of engine is also a less difficult problem than the cover, as it is a perfectly symmetrical casting at the region of the combustion chamber, where the maximum temperatures are encountered. This remark, however, does not apply to liners of the opposed piston type of engine where holes have to be cut in the combustion space, as in this case very serious stresses are introduced and several cases of cracked liners have occurred.

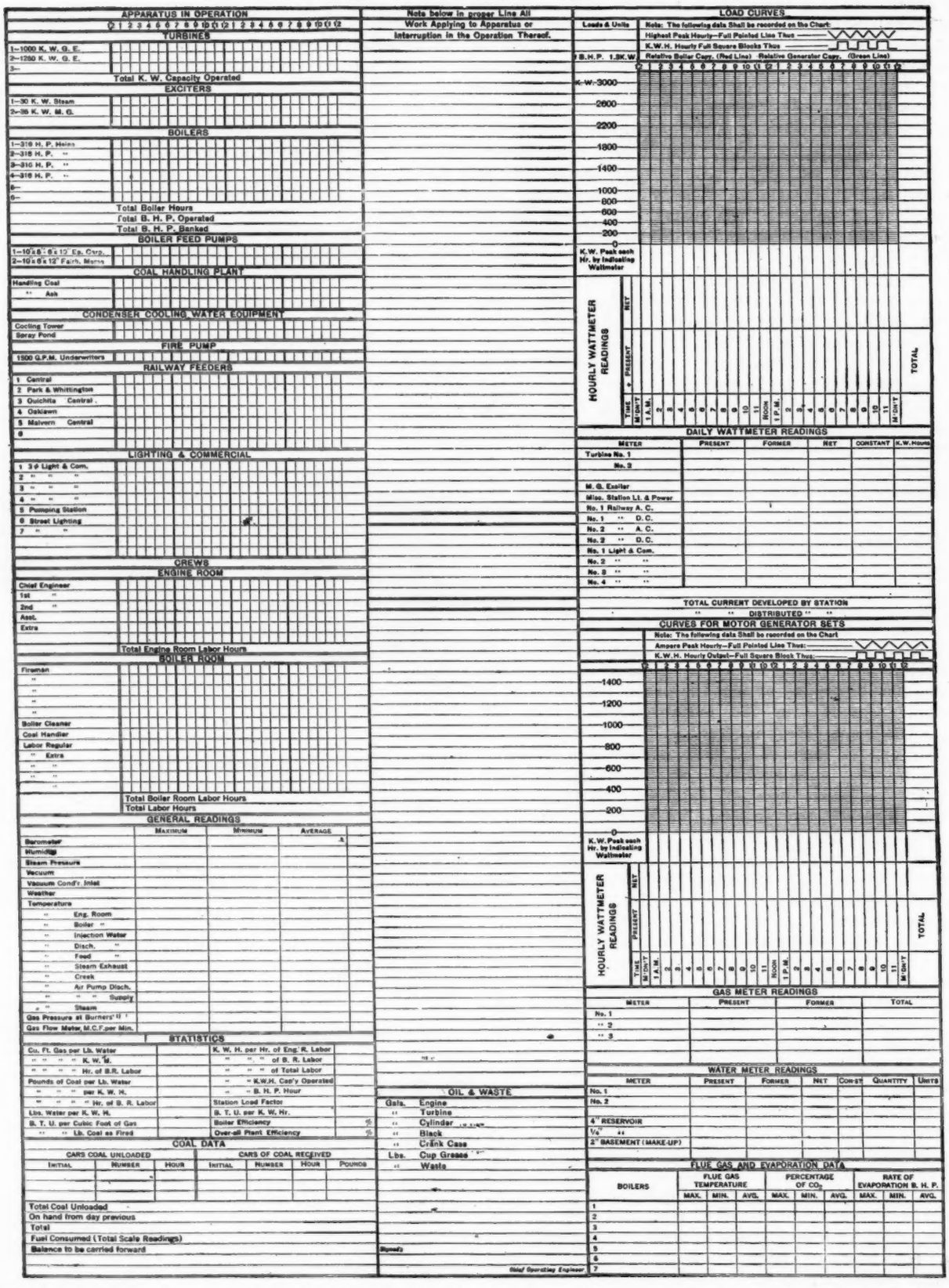


TWO- AND FOUR-STROKE-CYCLE CYLINDER HEADS

The freedom from failure due to heat stress in the port scavenging two-stroke-cycle engine is primarily attributable to the design of the cylinder cover which was evolved after careful investigation of the causes of failure on both two-stroke-cycle and four-stroke-cycle cylinder heads. The illustration shows at the right a four-stroke-cycle head with the quite common fracture between the valve cavities. It will be seen that the two-stroke-cycle cylinder to the left is less likely to have such fractures.

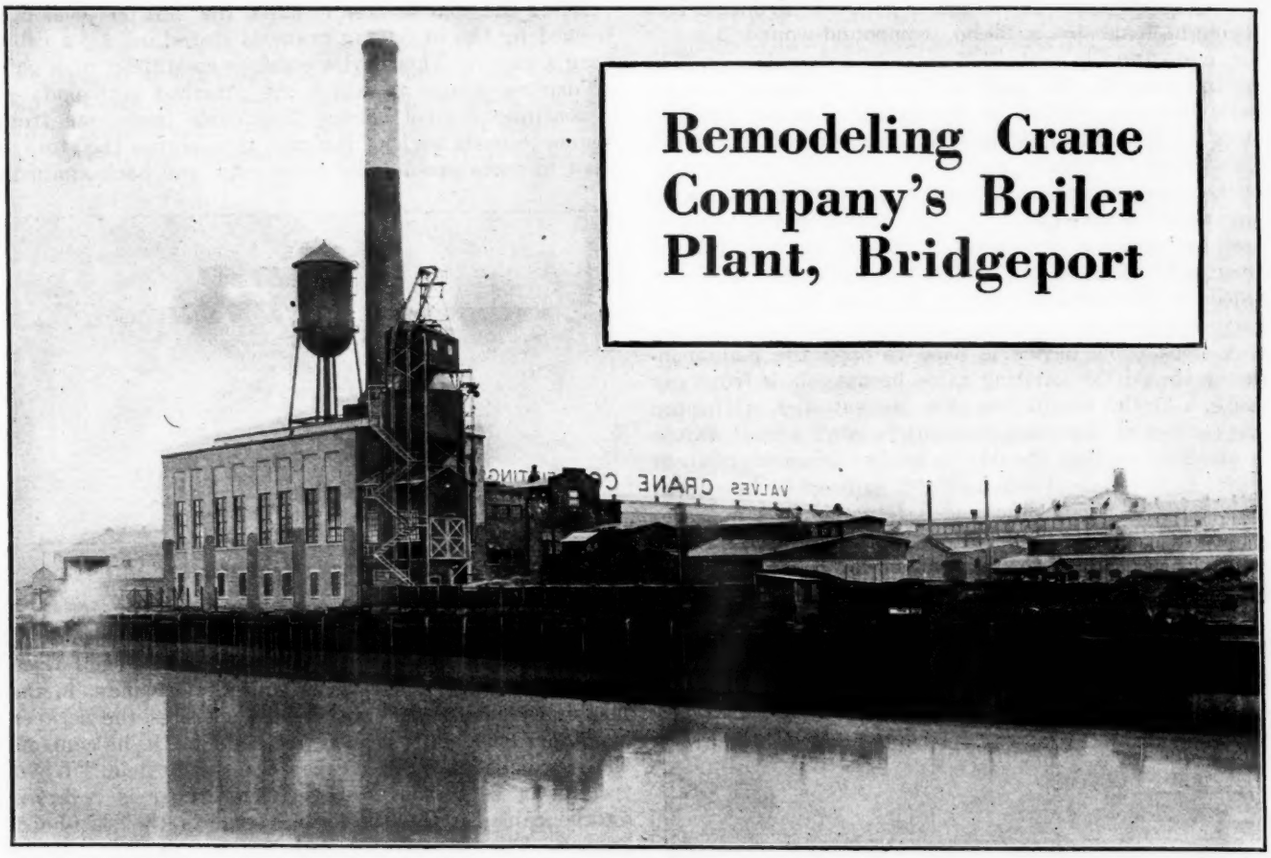
*Extract of paper read before the Institute of Marine Engineers.

A Complete Daily Log Sheet



Those in charge of power plants often find it difficult to arrange record forms that will be adequate yet convenient to keep and easy to interpret. Above is a reproduction of the daily log employed by the Citizens Electric Co., of Hot Springs, Ark. When filled out, it gives a complete record of the plant's operation for that day. With modifications to meet different conditions it might serve as a model for other plants.

Remodeling Crane Company's Boiler Plant, Bridgeport



THE boiler plant at the Crane Company's Bridgeport Works originally contained eight 3,500-sq.ft. hand-fired water-tube boilers, set at the ground level. The coal was wheeled to the furnaces in cars and the ashes wheeled out in barrows.

About May, 1921, the plant management realized the need of more steam and considered installing two 5,000-sq.ft. boilers alongside of the existing ones, extending the building to suit. At this time, however, the Government was offering power-plant equipment for sale, and the company's main office at Chicago purchased four boilers, stokers and other necessary equipment to rebuild the present plant. A new building became necessary at once as the newly purchased boilers were too large to fit into the existing boiler house.

The boilers were of the Stirling type, 8,230 sq.ft. each, set in batteries of two and were equipped with Westinghouse 8-retort stokers. In order to provide for proper ash disposal without expensive tunneling, the boiler-room floor was made 15 ft. above the ground line. The coal bunker was located outside so that the boiler house could be built of standard building construction and the firing aisle would be clean, light, well ventilated and unobstructed by overhead bunkers.

The ash hoppers under the stokers are arranged to carry a 16-hour run of ashes and are of steel-plate construction lined with hard-burned red brick. The air ducts to the stokers are steel plate, air being supplied by a 45,000-c.f.m. turbine-driven fan. The four stokers are driven from a common lineshaft by a single steam engine.

Steam pumps are used to handle the feed water. This water is supplied by the city and first passes through an open heater before going to the boilers. All this auxiliary equipment, such as stoker-drive engines, pumps, heater, etc., is in the basement, leaving the boiler room clear of everything but the boilers and stokers.

In order to take care of the extra draft required by the new boilers, the existing radial-brick stack was increased from 150 ft. to 200 ft. high. The addition to the stack shows plainly in the headpiece. Lined plate steel breechings carry the gases to the stack.

The generating equipment will remain as it is, no changes being contemplated along this line.

Coal-Handling Equipment

Coal for the plant is received in barges from Virginia and is unloaded by a locomotive crane to a hopper set flush with the ground line. From this hopper the coal is fed by an automatic loader (Fig. 2) to a high-speed skip hoist having a capacity of 40 tons of coal per hour. This device consists of a square chute bolted to the outlet of the coal hopper. To it is suspended a pivoted chute which is opened by the descending skip bucket coming in contact with and pulling down an operating bale. This bale, which is of the spring-draft friction type, equalizes the pull on the chute and prevents undue shock to bucket and loader at the instant of contact. It is attached to the chute and is closed when the skip bucket ascends. Counterweights are attached to assist in closing and to keep it closed.

Should the skip bucket fall by accident, the greater impact of the bucket causes the bale to swing back out of its path, preventing any damage to the loader.

The skip bucket is of the counterweighted type and is driven by an automatic electric hoist (Fig. 1). This

machine consists of a spiral-grooved drum driven through gears by a 20-hp. compound-wound 220-volt d.c. motor and is controlled entirely automatically. The control panel is also seen in Fig. 1. A shoe type magnetic brake is applied to the extended motor shaft to stop the machine quickly when the current is cut off. This occurs normally at the travel limits of the skip bucket; and in case of accidental current failure, at any point in the cycle. It also prevents the winding machine running backward in case of current failure. Overload and no-voltage release switches are also provided to cut out the motor in case of overload or current failure.

A slack cable device is used to open the main contactor should the hoisting cable become slack from any cause. At the top of the skip runway and at the top and bottom of the counterweight runway a limit switch is provided so that should the bucket or counterweight travel past its designed limits, it will open the switch and stop the hoist. Inclosed safety switches are also provided at the top and bottom of the runway, so that workmen can cut off the current while working at these points.

In the operation of the skip hoist the crane operator pushes a starting button and the bucket pumps up and down until the "stop" button is pushed. This

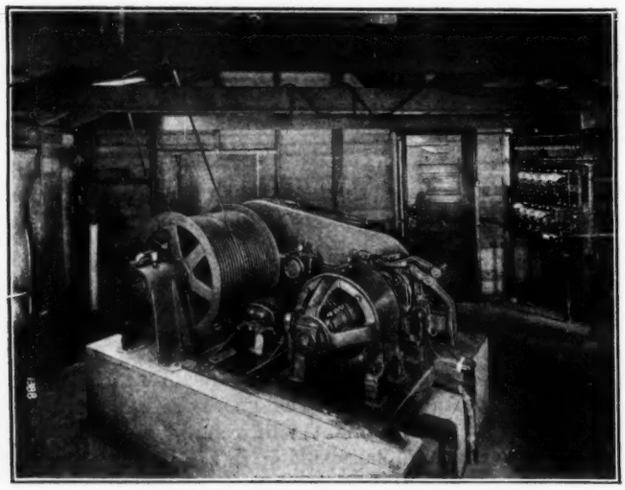


FIG. 1. AUTOMATIC ELECTRIC HOIST DRIVE AND CONTROL

hoist delivers the coal to a 30 x 30-in. double-roll crusher located over the coal bunker. After it is crushed, the coal falls into a 250-ton circular steel bunker, this capacity being sufficient for two days' run. The bunker is 25 ft. in diameter by 14 ft. high in the cylinder and is cone-shaped top and bottom.

From the bunker the coal is delivered to the stokers by a motor-driven coal-weighing larry running on tracks in front of the boilers. Extension hoppers are bolted to the stoker magazines to increase their capacity. This enables the larry operator to pile up several hours' run over each stoker so that he can be released for other duties.

The ashes falling from the stoker dump plates into the hoppers previously mentioned, pass through pivoted gates into cars hauled by a steam locomotive. At present ashes are used to fill in around the plant, but may be hoisted to an overhead bunker by the coal skip hoist and disposed of through a chute to barges.

After the coal bunker is filled, the "initial" coal pile formed by the unloading crane is stored out by a cable drag scraper. This device consists essentially of a pair of driving drums to which are attached both ends of a continuous steel cable. The cable leads out from drums to posts back of the pile, then across the storage yard to posts around the outer edge and back again to

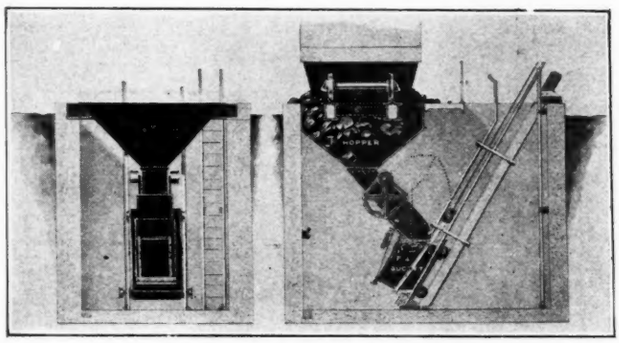


FIG. 2. AUTOMATIC LOADER FOR SKIP BUCKET

the front posts, then to the drums. On this cable is attached a steel scraper which stores the coal out from the "initial" pile and returns empty. One man, in the machinery house over the drums, operates the scraper system. By pushing a lever back and forth, he controls the movements of the scraper across the field. When reclaiming coal from storage, the scraper is reversed on the cable so that it scrapes coal to the skip-hoist hopper. The rate of storage is 60 tons an hour. Driving drums are propelled by a 25-hp. motor.

The scraper drive consists of a pair of 18-in. diameter cast-iron drums, with large friction wheels, both drum and wheels being cast in one piece. Between the drums is located the driving shaft, connected by a belt drive to the motor. On this shaft is a fiber friction pinion, which engages the large drum wheels.

The drum shafts are each mounted in spherical ball and socket bearings on one side, the other end being supported by an eccentric bearing, the arms of which are connected to the single control lever by rods.

When in operation, the motor runs continuously, the control lever being in neutral when vertical. Throwing the lever forward causes the rear drum friction wheel to engage the fiber friction pinion, the other drum running idle, its momentum being imparted by the unwinding cable. To prevent this drum gaining in speed, automatic counterweighted brakes are used. The machine is reversed by throwing the control lever in the opposite direction, when the brake will be automatically applied to the other drum.

It was foreseen by the company that with the boilers set 15 ft. above grade, it would be impossible to get coal to them until the coal-handling installation was complete. As the boilers and stokers were already on the site, it was early seen that there would be a lapse of a month or so after the boilers and stokers were set before the coal-handling equipment would be ready for operation. To provide for this emergency, a fuel-oil storage tank was located underground and the oil pumped to the boilers. Feed pipes were tapped through each boiler bridge wall and the oil sprayed into the furnaces by Babcock & Wilcox steam nozzles, three nozzles per furnace (Fig. 3). While the oil-burning system was more expensive to operate than coal burning, it

avoided the use of an expensive temporary coal-handling system.

The force required to operate the old boiler plant consisted of two firemen, one water tender and two coal and ash men, or five men per eight-hour shift. As the plant operated continuously for twenty-four hours each day, this meant fifteen men per day. It is estimated

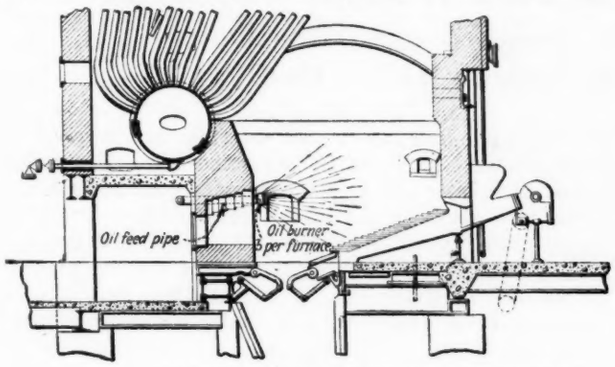


FIG. 3. TEMPORARY OIL-BURNING EQUIPMENT

that the new force will consist of only three men in the boiler room on each shift. This represents a saving of six men per day.

The engineering and construction supervision of the plant was done by the Crane Company engineers; the coal- and ash-handling system was designed and erected by the R. H. Beaumont Co., Philadelphia, Pa.

Why a Watt-Hour Meter Register Constant Is Used

BY MALCOLM MCKENZIE

A constant is just what the name implies; namely, a definite unchanging quantity. In a watt-hour meter there are two constants—the register constant and the calibrating constant. The calibrating, or test, constant is one used in testing the meter, but it is the purpose here to explain the register constant, which is often called the "multiplier." The "multiplier" designates a quantity by which the meter reading must be multiplied to ascertain the correct amount of energy used in a given circuit. However, the question is, How is this constant derived and why should it have different values in different meters?

To understand thoroughly the origin of the multiplying constant, one must know something of the principles on which the meter is built. Every watt-hour meter has a revolving element which is similar to the revolving element in a motor except that it is much lighter in weight. Every time the revolving element makes a revolution, a definite amount of energy has been expanded in the external circuit. Thus, it is readily seen that the only thing necessary to compute the energy consumed in the circuit is a counter or integrating speedometer placed on the shaft of the meter's moving element so as to record its revolutions in terms of kilowatt-hours. This is just what the register is. In some cases the register gives a direct reading in kilowatt-hours, while in other meters a multiplier is used.

There are two reasons for using a constant. One is that when a register is built, it is constructed to conform with certain characteristics of the meter on which it goes, such as capacity, type and speed. Now, when any of these characteristics are changed, as would be the case if the meter were rebuilt from one capacity to another, a new register would have to be used or the old one given a constant or multiplier that would give the correct reading when the meter reading was multiplied by it.

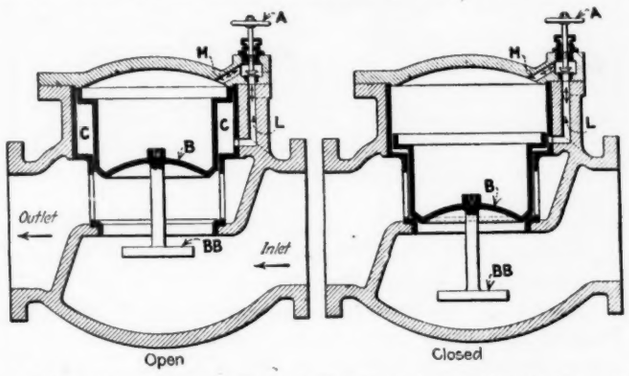
Practically all modern watt-hour meters have a four-dial gear train; that is, there are four hands on the register. Each of these dials is divided into ten divisions, hence the maximum reading obtainable is 9,999 kw.-hr. One more kilowatt-hour passed through the meter would leave the reading 0000. It is evident, then, that if a meter like this were put on a circuit using 35,000 or 40,000 kw.-hr. per month, it would be necessary to read it at least once a week to prevent the record from becoming lost. Therefore the register must be geared down where it will register only a small portion of the power used in the circuit, such as 3,500 or 4,000, and the register given a multiplier of, say, 10. Now when the meter indicates 750, the total energy supplied to the circuit is $750 \times 10 = 7{,}500$ kilowatt-hours.

Golden-Anderson Double Cushioning Check Valve

With a view to eliminating sudden shocks in pipe lines a check valve with a double cushioning feature has recently been developed by the Golden-Anderson Co., Pittsburgh, Pa. This valve is shown in the illustration in both open and closed positions.

The cushioning effect is produced in both the opening and closing operation of the valve. When opening, the fluid above the piston *B* flows through the ports *M* and *L* and valve *A* to the chamber *C*; the extent to which valve *A* is opened controls the speed at which the valve will function. This can be adjusted to suit the requirements of the service.

In closing, when a reversal of flow takes place a pressure is exerted on the piston *BB* which forces the



A DOUBLE CUSHIONED CHECK VALVE DESIGNED TO ELIMINATE WATER HAMMER AND SUDDEN SHOCKS FROM PIPE LINES

valve to its seat, the fluid in the cushioning chamber *C* flowing back again, to above the piston *B*. In following out this operation, it is readily seen that the valve is effectively cushioned at any point in the operation, thereby eliminating the possibility of disastrous water hammer due to sudden closing.

This valve is made in both angle and globe patterns and sizes up to 30 in. and is especially adapted for use in discharge lines of pumps and hydraulic-elevator service.

Three 70,000 Hp. Hydraulic Turbines To Be Installed at Niagara Falls

Largest Hydraulic-Power Tunnel in the World To Supply Water to the Three Largest Hydro-Electric Units Designed up to This Time—These Machines Will Operate Under a 215-ft. Head

NIAGARA FALLS is to have another record in hydro-electric units. When the three 37,500-hp. turbines were put in operation in 1919 and 1920 in an extension to Hydraulic Plant No. 3 of the Niagara Falls Power Co. (described in *Power*, Sept. 14, 1920), these were the largest machines in the world. Since that time the record has passed to the Canadian side in the five 55,000-hp. machines being installed at Queenston by the Hydro-Electric Power Commission of Ontario. This record is again to come back to the American side in three 70,000-hp. machines in another extension to Hydraulic Plant No. 3.

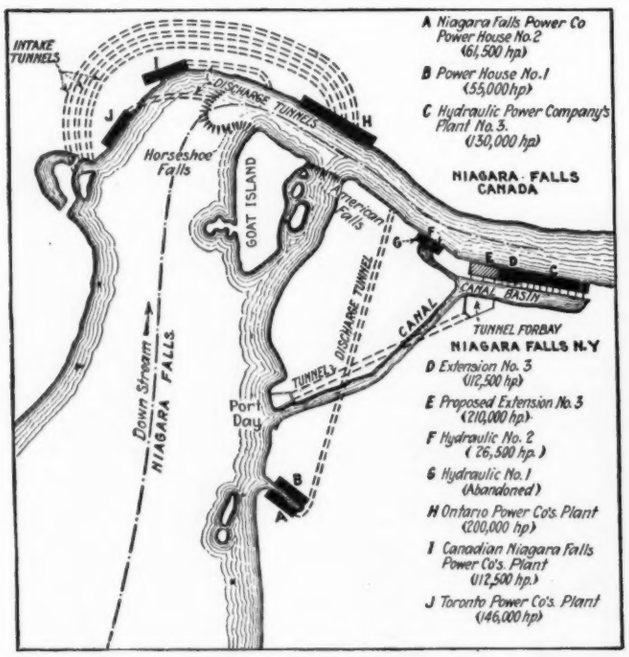

INTAKE TUNNELS
DISCHARGE TUNNELS
Horseshoe Falls
GOAT ISLAND
American Falls
NIAGARA FALLS CANADA
CANAL BASIN
TUNNEL FORBAY
NIAGARA FALLS N.Y
DISCHARGE TUNNEL
CANAL
TUNNEL
Down Stream
NIAGARA FALLS
Port Day
A Niagara Falls Power Co Power House No. 2 (61,500 hp)
B Power House No. 1 (55,000 hp)
C Hydraulic Power Company's Plant No. 3. (130,000 hp)
D Extension No. 3 (112,500 hp)
E Proposed Extension No. 3 (210,000 hp)
F Hydraulic No. 2 (26,500 hp.)
G Hydraulic No. 1 (Abandoned)
H Ontario Power Co's. Plant (200,000 hp)
I Canadian Niagara Falls Power Co's. Plant (112,500 hp.)
J Toronto Power Co's. Plant (146,000 hp)

FIG. 1. LOCATION OF PLANTS AND THEIR CAPACITIES

Large-scale water-power development at Niagara Falls on the American side has been done at two locations—one at *A* and *B* above the falls, Fig. 1, and the other at *C* and *D* below the falls. The former is where the first large development was attempted—the installation of 5,000-hp. units to operate under 140 ft. head, an engineering undertaking that attracted international interest at the time when it was undertaken. The water is conducted through penstocks from the forebay to the turbines located in a wheelpit about 140 ft. below the generators. A discharge tunnel carries the water from the turbines to the river below the falls. In these plants about 11 hp. is developed per cubic-foot-second of water delivered to the turbines.

In the other development the water is taken through a canal 100 ft. wide and 14-ft. deep, running from Port Day about one-half mile above the falls to a forebay at the top of the cliff, 4,400 ft. below the falls. From this forebay the water passes through penstocks to turbines located in a plant on the river's bank. These turbines operate under an effective head of 215 ft. and develop 22 hp. per cu.ft.-sec., or double that of the old plants.

When the extension to Hydraulic Plant No. 3 was finished and the three 37,500-hp. units put in operation, all the water allowed for diversion to power purposes, (19,500 cu.ft.-sec.) was utilized. Of this total water about 8,600 cu.ft. is being used in the wheelpit plants and 10,900 in the plant supplied through the canal.

To utilize the total allowed diversion more efficiently, the company has decided to use the wheelpit plants for stand-by service and divert this water to three 70,000-hp. turbines to be installed in another extension to the plant on the lower river's bank. This new arrangement will produce double the amount of power now being developed from the water. When this extension is completed, there will be installed in one building 16 units having a normal rating of 452,500 hp. capacity.

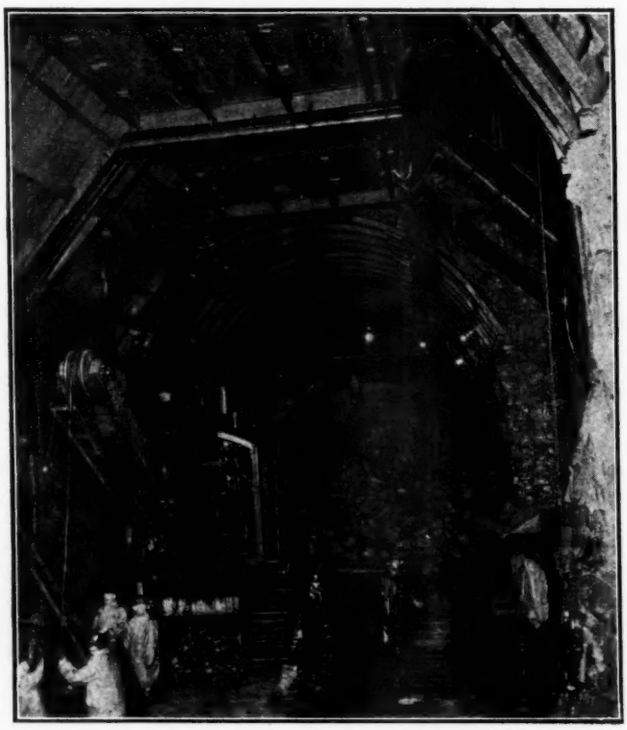


FIG. 2. INTERIOR OF NIAGARA FALLS POWER TUNNEL

The canal serving Hydraulic Plant No. 3 and its present extension has capacity only sufficient to serve the present installation of 242,500 hp., so that when the new extension was decided upon means had to be con-

sidered for conducting the additional water, 8,600 cu.ft.-sec., to the plant. Two methods of doing this were considered—first, an enlargement of the canal; second, a pressure tunnel. To enlarge the canal would have required additional property for lateral expansion or extensive dredging operations. After a careful analysis of the two propositions, it was decided that a pressure tunnel would be the more economical.

The route that the tunnel follows is indicated in

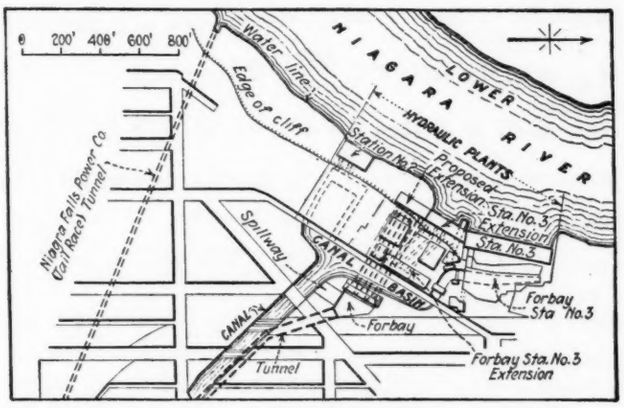


FIG. 3. LAYOUT AT DISCHARGE END OF TUNNEL

Fig. 1. Starting at an intake chamber at Port Day, it extends for about half its length on the up-river side of the present canal and then crosses under the canal and terminates in an open forebay adjacent to the present canal basin. Both forebays will be connected by a spillway (see Fig. 3) so that the joint forebay can be filled from either canal or tunnel. The tunnel is cut in solid rock and will, when lined with concrete, be of a horseshoe shape having maximum horizontal and vertical dimensions of 32 ft., giving it a cross-sectional area of 875 sq.ft. As driven in the rock, the tunnel is approximately 36 ft. inside the rock-break line and is 4,300 ft. long. Fig. 2 is a view of the inside of the tunnel in the process of construction and gives an idea of its size, which is the largest in the world with the exception of the ship tunnel for the Rove Canal at Marseilles, France.

On each side of the intake is a concrete side wall, and between the walls there will be four concrete gate piers, having slots in their sides, into which can be placed shut-off gates should it ever be desired to unwater the tunnel. The forebay will be connected to the power house through three 21-ft. concrete-lined penstocks, each of which will supply water to a 70,000-hp. vertical-shaft turbine. Across the forebay is a gate-house controlling the entrance to the three penstocks. For about 80 ft. at their lower ends, the penstocks will have riveted-steel-plate linings inside the concrete which will act as anchors for the penstock valves and turbine casings to the penstocks.

The turbines will be of the vertical-shaft Francis reaction type similar to the 37,500-hp. machines now installed in extension No. 3, the general proposed arrangement being indicated in Fig. 4.

As announced in the N. E. L. A. 1922 electrical Apparatus Committee Report, an unusual combination of auxiliary and excitation supply for these units is proposed. Each of the generators, which will be of approximately 52,000 kw. capacity, is to include a small alternating-current generator on the shaft of the main unit. This auxiliary generator will be of sufficient capacity to supply all the auxiliaries for the unit, including the motor-driven exciter set, governor pump, ventilation fan and any other equipment that may be required. All the auxiliaries will be arranged for starting from the alternating-current service bus. The motor-driven exciter set will include two exciters, one for the main unit and a smaller one for exciting the fields of the auxiliary generator. After the exciter set and the main unit have both been started and the auxiliary generator excited, the auxiliary alternating-current supply will be transferred from the service-bus feeder to the auxiliary generator. With the very large capacity of the main units involved, it is considered advisable that they should be made as complete individual power plants in so far as possible.

The construction was started in the spring of 1921. At present the intake work is well along toward completion, the tunnel is 45 per cent and the penstocks 35 per cent completed.

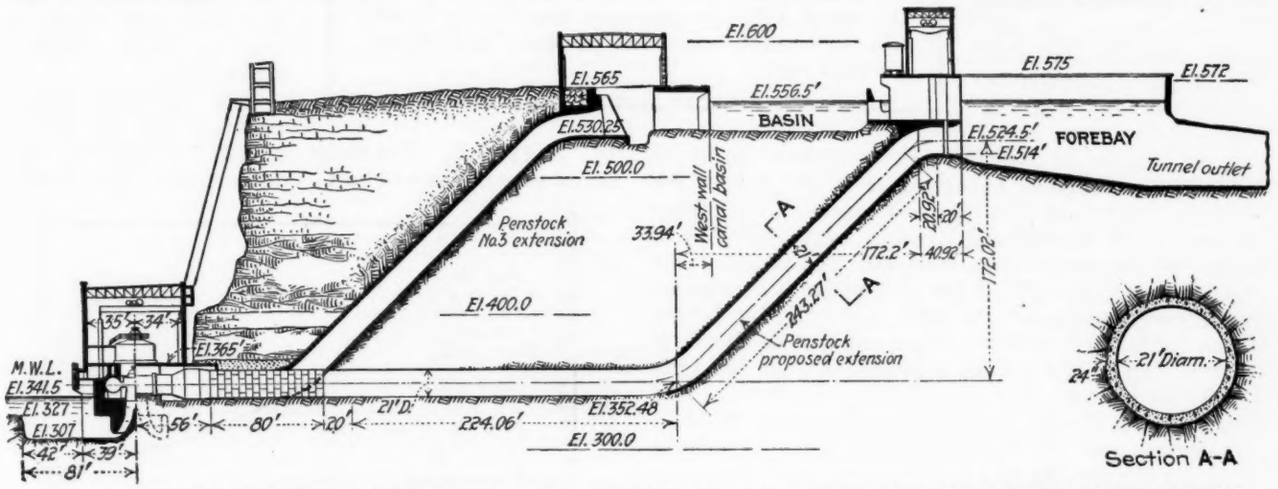


FIG. 4. SECTION THROUGH POWER HOUSE AND ONE PENSTOCK FOR EXTENSION NOW UNDER CONSTRUCTION

The new work at Niagara Falls is being carried on by the Niagara Falls Power Co., of which John L. Harper is vice president and chief engineer. Construction work is under the direction of O. D. Dales, construction engineer, and the contract for the entire construction is held by the Read-Coddington Engineering Co., of Niagara Falls.

Finding the Mean Effective Pressure of an Engine by Computation

IT IS often necessary to calculate the mean effective pressure that will result from using steam at some definite initial pressure and expanding it a given number of volumes in the engine cylinder.

It has been shown in previous articles that the mean effective pressure is proportional to the area of the indicator diagram. If a diagram has twice the area of another having the same length and spring scale, the m.e.p. will be twice as much as in the second diagram. If the initial pressure continued throughout the stroke, the diagram would be a rectangle like *AFDH*, Fig. 1. If cutoff took place at *B* and if there were no clearance and no back pressure, the expansion would follow the curve *BC* and the diagram would be *ABCDH*. The diagram *ABCDH* is, as shown in Fig. 2, made up of two parts, one of which is *ABJH* with a height equal to *AH*. The rectangular hyperbola curve *BC* defines the height of the second part of the diagram *BCDJ*. Since for any initial volume and pressure represented by *B* the hyperbolic curve has a fixed position, it is obvious that the average height of the area under it of a given length must bear a definite relation to *AH*, the height of the initial area *ABJH*. If the initial condition *B* (that is, the initial volume and pressure, or the area *ABJH*) be altered, the path of the curve changes and the relation of the average height of the area *BCDJ* to *AH* changes, but the relation now existing is definite and depends upon the ratio of expansion *R* or the ratio of *HD* to *HJ*. For example, if the number of expansions, or the ratio of the volume at the end of expansion at *C* to the initial volume at *B*, be 4, then the ratio of the area *BCDJ* to *ABJH* is 1.3863, or *BCDJ* is 1.3863 times as large as *ABJH*.

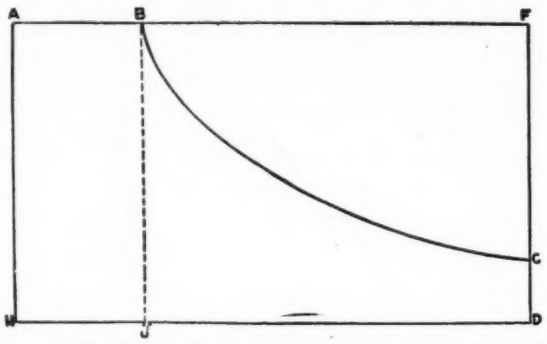


FIG. 1. EXPANSION IN CYLINDER WITHOUT CLEARANCE

This article discusses the method of computing the mean effective pressure due to any condition of initial and back pressures and to any point of cutoff.

This ratio or relation existing between the areas before and after the point of cutoff is the hyperbolic logarithm of the ratio of expansion *R*. In Table I are given, under the column marked "Logarithm," this relation for various expansion ratios.

The area *BCDJ* will be the hyperbolic logarithm of *R* times the area *ABJH* and the sum of the two areas will be 1 + hyp. log *R* times the initial area *ABJH*. If we set the total length of the diagram as equal to unity, then the length *HJ* of the area *ABJH* will be $\frac{\text{unity}}{\text{ratio of expansions}}$, or $\frac{1}{R}$, since the ratio *R* is $\frac{HD}{HJ}$. If the height *AH* of the area *ABJH* is *P*, then the area of *ABJH* is $P \times \frac{1}{R}$, or pressure × volume = $\frac{P}{R}$. It has been shown that the entire area *ABCDH* is (1 + hyp. log *R*) times the area *ABJH*, or *ABCDH* = (1 + hyp. log *R*.) × $\frac{P}{R}$ = $P\frac{(1 + \text{hyp. log } R)}{R}$. In this the diagram length has been taken as unity (1) so that the average height or average pressure will be the area divided by the length or since its length is unity, $PM = P\frac{(1 + \text{hyp. log } R)}{R}$.

The mean effective pressure in a diagram with no clearance and no back pressure may then be found as follows:

Rule. Add 1 to the hyperbolic logarithm of the ratio of expansion; multiply the sum by the absolute initial pressure, and divide by the ratio of expansions. The quotient will be the mean forward absolute pressure represented by the ideal diagram.

Example. What will be the ideal mean forward pressure with steam at 100 lb. gage, cutoff ¼ stroke?

The absolute initial pressure will be 100 + 14.7 = 114.7 lb. The cutoff being at ¼ stroke, the number of expansions will be 4. The hyperbolic logarithm of

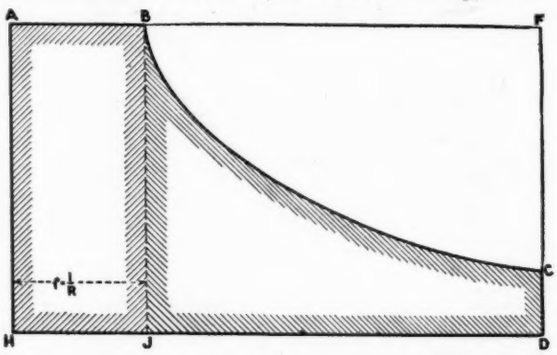


FIG. 2. WORK DIAGRAM OF NO-CLEARANCE ENGINE

4 is (Table I.). 1.3863. Using the rule, $P_m = \frac{(1 + 1.3863)}{1.3863}\ 114.7 = 68.427$ lb.

This is the mean forward pressure measured from the vacuum or zero pressure line *HD* and is the mean effective pressure if there is no back pressure, bringing *HD* above the zero line. However, it is impossible to obtain

a perfect vacuum in the exhaust line, and some back pressure must exist. This will range from 3.5 for condensing and 15 to 18 lb. for non-condensing operations. This pressure continues practically uniform for the entire return stroke in a four-valved engine and must be deducted from the mean pressure to give the mean effective pressure in the cylinder.

TABLE I. HYPERBOLIC LOGARITHMS OF RATIOS OF EXPANSION

Ratio	Logarithm	Ratio	Logarithm	Ratio	Logarithm	Ratio	Logarithm
1	0	3.33	1.2060	5.5	1.704	8	2.079
1.25	0.2231	3.5	1.252	5.75	1.749	8.25	2.110
1.5	0.4054	3.75	1.321	6	1.791	8.5	2.140
1.75	0.5596	4	1.386	6.25	1.832	8.75	2.169
2	0.6931	4.25	1.446	6.5	1.871	9	2.197
2.25	0.8109	4.33	1.4656	6.75	1.909	9.5	2.251
2.5	0.9162	4.5	1.504	7	1.945	10	2.302
2.75	1.0116	4.75	1.558	7.25	1.981	12	2.484
3	1.0986	5	1.609	7.5	2.014	15	2.708
3.25	1.1786	5.25	1.658	7.75	2.047	18	2.890

Effect of Clearance

The real ratio of expansion is always less than that indicated by the point of cutoff on account of the clearance. In Fig. 3 is a diagram showing cutoff at ¼ stroke, giving an apparent ratio of expansion of 4. Suppose that the clearance was 10 per cent of the stroke volume. The volume filled with steam up to cutoff at B is not AB but $A'B$. The expansion will not follow the line BC, but since there is a large volume $A'B$ of steam at initial pressure to expand, the curve will be BC'. This curve is a hyperbolic curve as is BC, but has a higher pressure at all points in the piston stroke for the product of the initial pressure and volume is greater, due to the larger volume $A'B$. The volume at the end of the stroke at D is $OH + HD$ instead of HD as in case of no clearance. The real ratio of expansion is, then, $\frac{OH + HD}{A'B}$

If we call the stroke volume HD unity, then OH, since it is 10 per cent of the stroke volume, is 0.1. AB is, as

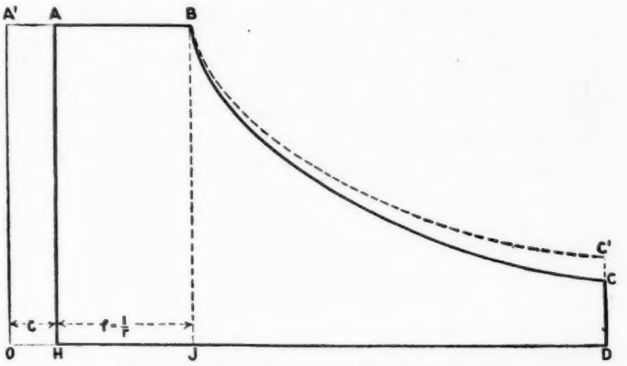


FIG. 3. EFFECT OF CLEARANCE ON THE EXPANSION

has been stated, ¼ of HD or 0.25. The real ratio of expansion is then $\frac{OH + HD}{A'B} = \frac{0.1 + 1}{0.25 + 0.1} = \frac{1.1}{0.35} =$ 3.14 and not 4. The *real ratio* of expansion is the total volume of the cylinder including the clearance volume divided by the volume up to cutoff including the clearance volume. Calling the per cent of stroke to cutoff f, and the apparent expansion r which is $\frac{1}{f}$, and the percentage of clearance c while the stroke volume is unity

TABLE II. MEAN FORWARD PRESSURE PER POUND OF INITIAL, WITH DIFFERENT CLEARANCES AND POINTS OF CUTOFF

Fraction of Stroke Complete at Cut-off.		Percentage of Clearance												Percentage of Clearance											Fraction of Stroke Complete at Cut-off.	
		0%	1%	1.5%	2%	2.5%	3%	3.5%	4%	4.5%	5%	5.5%	6%	6.5%	7%	7.5%	8%	8.5%	9%	9.5%	10%	10.5%	11%	11.5%	12%		
1/10	.1	3303	.3439	.3505	.3568	.3630	.3690	.3750	.3808	.6864	.3919	.3974	.4027	.4076	.4126	.4176	.4225	.4271	.4320	.4366	.4409	.4453	.4498	.4540	.4580	1/10	.1
1/9	.111	.3549	.3677	.3738	.3798	.3856	.3913	.3968	.4022	.4075	.4129	.4178	.4227	.4278	.4326	.4373	.4417	.4462	.4507	.4549	.4593	.4633	.4676	.4715	.4757	1/9	.111
1/8	.125	.3849	.3966	.4023	.4078	.4132	.4187	.4237	.4287	.4338	.4386	.4433	.4480	.4527	.4571	.4615	.4657	.4700	.4740	.4782	.4821	.4858	.4897	.4938	.4973	1/8	.125
1/7	.143	.4213	.4320	.4370	.4420	.4471	.4518	.4565	.4612	.4655	.4699	.4743	.4788	.4827	.4871	.4908	.4951	.4987	.5026	.5062	.5101	.5138	.5173	.5205	.5242	1/7	.143
	.15	.4346	.4447	.4497	.4546	.4794	.4639	.4684	.4729	.4774	.4816	.4860	.4901	.4939	.4978	.5020	.5059	.5096	.5131	.5169	.5204	.5237	.5274	.5309	.5342		.15
1/6	.167	.4662	.4757	.4802	.4844	.4890	.4933	.4973	.5014	.5056	.5096	.5134	.5173	.5210	.5245	.5283	.5318	.5352	.5389	.5417	.5457	.5488	.5517	.5551	.5583	1/6	.167
3/16	.188	.5013	.5097	.5138	.5181	.5217	.5259	.5295	.5332	.5367	.5405	.5440	.5474	.5511	.5546	.5579	.5610	.5639	.5673	.5705	.5736	.5764	.5791	.5825	.5848	3/16	188
1/5	.20	.5219	.5298	.5336	.5376	.5414	.5449	.5482	.5517	.5556	.5588	.5623	.5656	.5687	.5716	.5750	.5782	.5812	.5841	.5868	.5901	.5924	.5954	.5982	.6009	1/5	.20
	.21	.5376	.5453	.5489	.5523	.5560	.5595	.5628	.5664	.5698	.5730	.5760	.5795	.5821	.5853	.5882	.5910	.5944	.5968	.5998	.6028	.6055	.6081	.6106	.6129		.21
	.22	.5533	.5602	.5639	.5673	.5704	.5740	.5773	.5804	.5834	.5868	.5900	.5931	.5959	.5986	.6011	.6043	.6073	.6101	.6128	.6154	.6177	.6199	.6230	.6249		.22
	.23	.5681	.5750	.5781	.5815	.5848	.5878	.5913	.5940	.5971	.6001	.6029	.6063	.6087	.6118	.6138	.6166	.6192	.6225	.6248	.6270	.6300	.6318	.6345	.6371		.23
	.24	.5827	.5891	.5922	.5952	.5986	.6012	.6042	.6071	.6106	.6131	.6162	.6184	.6212	.6239	.6264	.6297	.6319	.6340	.6369	.6397	.6413	.6438	.6462	.6485		.24
1/4	.25	.5966	.6025	.6059	.6090	.6120	.6148	.6174	.6207	.6229	.6258	.6286	.6312	.6336	.6359	.6390	.6410	.6438	.6465	.6480	.6506	.6528	.6550	.6570	.6602	1/4	25
	.26	.6105	.6162	.6190	.6218	.6251	.6274	.6304	.6332	.6359	.6385	.6408	.6430	.6460	.6479	.6507	.6533	.6548	.6571	.6594	.6626	.6646	.6665	.6682	.6711		.26
	.27	.6232	.6294	.6319	.6350	.6370	.6398	.6424	.6448	.6480	.6501	.6531	.6549	.6576	.6601	.6626	.6649	.6670	.6691	.6710	.6728	.6757	.6772	.6799	.6812		.27
	.28	.6363	.6416	.6445	.6471	.6496	.6520	.6551	.6572	.6600	.6618	.6644	.6669	.6692	.6714	.6735	.6755	.6773	.6803	.6818	.6833	.6859	.6885	.6895	.6918		.28
	.29	.6491	.6545	.6569	.6592	.6613	.6642	.6660	.6686	.6712	.6736	.6759	.6780	.6800	.6819	.6849	.6865	.6880	.6906	.6919	.6943	.6967	.6990	.6997	.7018		.29
3/10	.30	.6609	.6663	.6684	.6712	.6729	.6755	.6779	.6803	.6825	.6845	.6864	.6882	.6911	.6927	.6954	.6966	.6991	.7002	.7024	.7046	.7067	.7087	.7107	.7125	3/10	.30
5/16	.313	.6760	.6805	.6830	.6855	.6878	.6899	.6919	.6938	.6956	.6985	.7000	.7026	.7039	.7063	.7073	.7096	.7117	.7138	.7157	.7176	.7194	.7211	.7226	.7241	5/16	.313
	.32	.6851	.6891	.6914	.6935	.6956	.6974	.7004	.7021	.7035	.7062	.7074	.7099	.7123	.7131	.7152	.7173	.7193	.7211	.7229	.7245	.7261	.7275	.7289	.7319		.32
1/3	.333	.6988	.7029	.7047	.7076	.7092	.7106	.7132	.7144	.7168	.7190	.7212	.7219	.7239	.7257	.7275	.7292	.7308	.7323	.7353	.7366	.7378	.7389	.7417	.7426	1/3	.333
	.34	.7067	.7115	.7130	.7145	.7171	.7183	.7207	.7230	.7238	.7259	.7279	.7298	.7316	.7333	.7349	.7364	.7378	.7391	.7421	.7432	.7442	.7469	.7477	.7484		.34
	.35	.7178	.7220	.7232	.7256	.7266	.7288	.7310	.7330	.7350	.7368	.7370	.7402	.7417	.7432	.7445	.7457	.7468	.7496	.7506	.7514	.7540	.7546	.7571	.7575		.35
	.36	.7281	.7316	.7338	.7346	.7367	.7386	.7405	.7422	.7439	.7454	.7468	.7482	.7511	.7523	.7533	.7543	.7569	.7577	.7602	.7608	.7632	.7636	.7658	.7660		.36
3/8	.375	.7433	.7458	.7476	.7494	.7510	.7525	.7539	.7569	.7582	.7593	.7603	.7630	.7639	.7646	.7671	.7676	.7700	.7711	.7725	.7739	.7752	.7766	.7779	.7792	3/8	.375
	.38	.7475	.7512	.7528	.7544	.7559	.7573	.7586	.7615	.7626	.7636	.7662	.7670	.7677	.7702	.7707	.7730	.7733	.7755	.7767	.7781	.7794	.7807	.7820	.7832		.38
	.39	.7566	.7613	.7627	.7640	.7653	.7664	.7691	.7700	.7708	.7734	.7740	.7764	.7768	.7791	.7793	.7815	.7824	.7837	.7850	.7862	.7875	.7888	.7900	.7912		.39
2/5	.40	.7665	.7691	.7719	.7729	.7738	.7765	.7772	.7778	.7802	.7806	.7829	.7831	.7853	.7874	.7880	.7892	.7905	.7918	.7930	.7942	.7954	.7966	.7978	.7990	2/5	.40
7/16	.438	.8000	.8024	.8030	.8044	.8063	.8068	.8079	.8096	.8104	.8115	.8127	.8138	.8149	.8161	.8172	.8182	.8193	.8204	.8214	.8224	.8235	.8244	.8254	.8264	7/16	.438
	.45	.8089	.8127	.8130	.8141	.8156	.8165	.8176	.8187	.8199	.8210	.8221	.8231	.8242	.8252	.8283	.8273	.8283	.8293	.8303	.8312	.8322	.8331	.8341	.8350		.45
1/2	.50	.8466	.8484	.8492	.8503	.8513	.8522	.8530	.8539	.8548	.8556	.8565	.8573	.8582	.8590	.8598	.8606	.8614	.8622	.8629	.8637	.8644	.8652	.8659	.8667	1/2	.50
	.55	.8733	.8792	.8810	.8817	.8824	.8831	.8838	.8844	.8851	.8858	.8864	.8871	.8877	.8883	.8890	.8896	.8902	.8908	.8914	.8920	.8925	.8931	.8937	.8942		.55
9/16	.563	.8868	.8875	.8882	.8888	.8895	.8901	.8908	.8914	.8920	.8926	.8932	.8938	.8944	.8950	.8956	.8962	.8968	.8973	.8979	.8984	.8989	.8995	.9000	.9005	9/16	.563
3/5	.60	.9064	.9076	.9081	.9087	.9092	.9097	.9102	.9107	.9112	.9117	.9122	.9127	.9132	.9136	.9141	.9146	.9150	.9155	.9159	.9164	.9168	.9173	.9177	.9181	3/5	.60
5/8	.625	.9188	.9194	.9201	.9206	.9210	.9215	.9220	.9224	.9228	.9233	.9237	.9241	.9245	.9249	.9253	.9257	.9261	.9265	.9269	.9272	.9276	.9280	.9284	.9288	5/8	.625
	.65	.9300	.9308	.9312	.9316	.9320	.9323	.9327	.9331	.9335	.9338	.9342	.9340	.9349	.9352	.9356	.9359	.9363	.9366	.9369	.9373	.9376	.9379	.9382	.9385		.65
2/3	.667	.9371	.9378	.9382	.9385	.9389	.9392	.9396	.9399	.9402	.9405	.9408	.9411	.9415	.9418	.9421	.9424	.9427	.9430	.9433	.9436	.9438	.9442	.9444	.9447	2/3	.667
11/16	.688	.9451	.9457	.9460	.9463	.9466	.9469	.9472	.9475	.9478	.9480	.9483	.9486	.9489	.9491	.9494	.9497	.9499	.9502	.9505	.9507	.9509	.9512	.9514	.9517	11/16	.688
7/10	.70	.9497	.9502	.9505	.9508	.9511	.9513	.9516	.9518	.9521	.9524	.9526	.9528	.9531	.9533	.9536	.9538	.9541	.9543	.9546	.9548	.9550	.9552	.9554	.9557	7/10	.70
3/4	.75	.9657	.9661	.9663	.9665	.9667	.9668	.9670	.9672	.9674	.9675	.9677	.9679	.9680	.9682	.9684	.9685	.9687	.9688	.9690	.9691	.9693	.9695	.9696	.9698	3/4	.75

or one hundred per cent, the *real* ratio of expansion R is

$$R = \frac{1 + c}{1 + c}$$

By substituting this value in the equation $P_m = P\frac{(1 + \text{hyp. log } R)}{R}$, we obtain the mean effective pressure. But this m.e.p. is for the entire diagram *A'BC'DO*. Since only that part of the diagram indicated by *ABCDH* represents work, the m.e.p. obtained if the foregoing formula for P_m is used, will be greater than the m.e.p. of the actual work diagram. If we had the area of the entire diagram including the clearance and from this area deducted the clearance, we would have the diagram area *ABCDH*. Dividing this area by the stroke volume would give us the m.e.p. of the work diagram.

The mean effective pressure, or the mean height of *A'BCDO* is P_m. The length is $1 + c$, giving the area as $P_m(1 + c)$. But $P_m = P\frac{(1 + \text{hyp. log } R)}{R}$, and by substituting the value of R and multiplying by $(1 + c)$ we have the whole area $= P(1 + \text{hyp. log } R)(f + c)$. Subtracting the area of the clearance, the area of the work diagram is $P[(1 + \log R)(f + c) - c]$. The mean effective pressure, since the length *DH* is unity, becomes $P_m = P[(1 + \log R)(f + c) - c]$.

Rule. To find the mean pressure of the ideal diagram corrected for clearance divide one, plus the clearance, by the clearance plus the fraction of the stroke up to cutoff. This gives the value of "R." Then multiply 1 plus the hyperbolic logarithm of "R" by sum of the fraction of the stroke up to cutoff and the clearance percentage. From the result subtract the clearance. This gives the mean forward pressure per pound of initial pressure for the ideal diagram corrected for the effect of clearance.

The m.e.p. per pound of initial pressure has been calculated for various cutoffs and for different percentages of clearance. These values appear in Table II.

From the forward pressure given in Table II must be deducted the average back pressure to arrive at the mean effective pressure. In the case of a single-valve engine the closing of the exhaust port fairly early in the stroke causes the average back pressure to be somewhat higher than the pressure existing in the exhaust line. For non-condensing operating with atmospheric exhaust, the average back pressure will be close to 18 lb. absolute. This value in calculating the m.e.p. should be deducted from the value given in Table II.

In the case of a Corliss engine compression begins later than in the single-valve engine and the average back pressure with atmospheric exhaust will be close to 17 lb. absolute. For condensing operation, with 26 in. vacuum, the pressure will be about 7 lb. absolute.

The unaflow when operating condensing starts the compression at 90 per cent full stroke. If the compression be along the PV = constant curve, the average back pressure with 26 in. vacuum will be approximately 10 lb. absolute, and for non-condensing operating with a later compression will be close to 20 lb. pressure.

Effect of Moisture in a Lubricating System

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EVERY circulatory oiling system is susceptible to contamination with moisture. The steam turbine leaks steam and water from its shaft stuffing boxes. The steam-engine cylinder blows steam and water through the piston- and valve-rod packings. No matter how good these packings are, water is going to get through some time, even if it does not continually leak a little, as is more likely to be the case.

Oil in inclosed crankcases of motor-driven air compressors, where there is no steam to leak, will absorb moisture from the air within the crankcases. When a warm crankcase cools off, moisture is sometimes condensed out of the cooled air, just as a gasoline storage tank will absorb moisture from the air.

Preventable Causes of Moisture

When cooling coils are used to lower the oil temperature, they will at times spring a leak. Steam coils used for heating oils in cold weather may leak and water collect in the oil. Washing down the engine-room floor has been the cause of many oil emulsions because the man on the end of the hose was not particular about splashing water into the crankpit of the engine. Barrels stored out of doors on end will collect water on the top when it rains or snows. The water soaks through the wood, dissolves the glue that is on the inside of the barrel to prevent the leakage of the oil, and the mixture of glue and water is absorbed by the oil.

Water in a circulatory system does a lot of things that do not improve lubrication. In the first place, if it becomes separated into small globules which are carried to the bearings with the oil, it is quite likely to cause a rupture of the oil film within the bearings. Unless the film is reformed at once, metal-to-metal contact will occur and the bearings will wear.

Furthermore, water is one of the finest things in the world to promote chemical reactions of all kinds. If there are any impurities in the oil, absorbed from the air, such as sulphur fumes around a blast furnace or iron foundry, or bits of metal, smoke, coal dust, or even road dust blown in through the window, moisture in the oil will be a great aid in causing trouble. Sulphur fumes coming in contact with dry oil will not cause trouble, but let there be a little moisture in the oil, and the sulphur gas at once combines with it, forming sulphuric acid. Ammonia vapor from leaks in an ammonia line will be absorbed very quickly by any moisture in the oil, with the formation of emulsions of ammonia and oil.

Again, when oil begins to break down, as all oils will, with long-continued exposure to heat and agitation with air, moisture will assist the chemical reactions accompanying the breaking down of the oil. Acidity may then develop, or emulsions may be formed with the constituents of the oil.

Even though there may be no trouble with the oil, the moisture may cause rusting of iron or steel parts.

Although it might be possible for corrosion to occur, due to acidity in the oil, it is quite unlikely to happen unless moisture is present. In certain tests that the writer has seen made on highly polished steel balls immersed in oil containing oleic acid in quantities varying from 0.1 to 5.0 per cent, there was no corrosion whatever in any case unless moisture was introduced by blowing air through the oil. Even then, when several sets were in series—that is to say, when the air was blown from one batch into the next—the only corrosion that took place occurred in the first batch, which absorbed all the moisture from the air. The steel balls in the following batches were in no way affected.

Formations of Emulsions

When oil that has been used for a long time, or even when fresh oil that has not been perfectly refined, is thoroughly agitated with water, as is the case in practically all circulatory systems, the moisture combines either mechanically or chemically with the impurities and forms an emulsion. In other words the water does not separate from the oil. The emulsion, an intimate mixture of oil and water, is a great collector of grit and foreign matter. As it carries these particles through the bearings, they act as abrasives and wear the metal away. It is quite likely that if all foreign matter could be kept out of the bearings and the oil film were properly formed and maintained, there would be practically no wear of bearings whatever.

Emulsions gradually increase in viscosity, collecting more water and more foreign matter, and eventually may become so thick that they will not flow through the oil lines leading to the bearing, unless forced under great pressure. When this happens, the bearing becomes hot and goes to pieces through lack of sufficient lubrication.

Aside from the many troubles directly laid to the door of moisture, there are those caused by sludges. Sludge may be caused by the separation of paraffin wax from an improperly refined oil or the decomposition of a perfectly refined oil under conditions of extremely high temperature and continued exposure to moisture and air. Decomposition of oil is usually accompanied by the development of petroleum acids, which, although practically neutral as far as their action on metals is concerned, are prone to form sludges when moisture and foreign material get into the system.

There is also the possibility of metallic soaps being formed from the products of decomposition, moisture, and the metal of the bearings, oil lines, coolers, etc. Sludges from this cause seem to be particularly prominent when copper or zinc is present in some part of the system. Sludge on cooler pipes acts as insulation, reduces the efficiency and prevents proper cooling of the oil. The hot oil decomposes further and the loss of oil is increased.

Most Trouble Caused by Moisture

It would seem from this discussion that moisture is at the bottom of most of our troubles from emulsions, deposits and sludges in the circulatory system. If we could eliminate the moisture, these troubles would be at an end—certainly, the greater part of them.

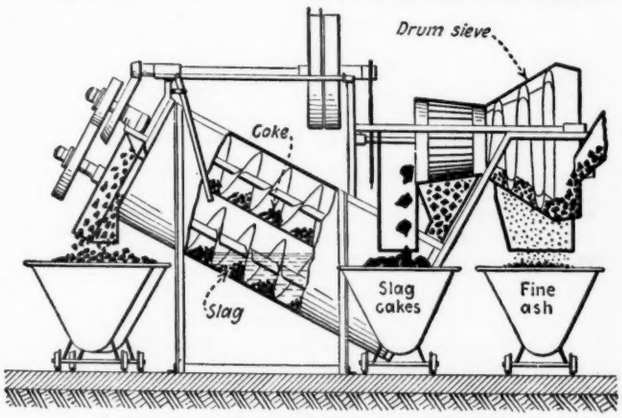
The first thing to do is to provide every means possible to prevent moisture from getting into the system. If it cannot be kept out one hundred per cent, then something should be done to remove it regularly and completely from the oil, so that there will be no accumulation of moisture. Separators and filters are valuable in this respect and are considered quite necessary accessories to most steam-engine and turbine installations.

Heat is advantageous in assisting in the removal of moisture and other contaminating matter from the oil because it reduces the viscosity. Water will separate more quickly from a thin oil than from a thick oil, from a hot oil than from a cold oil, and from a low-density oil than from a high-density oil.

A carefully refined oil, one that has been made especially to withstand the influences of moisture and other foreign matter, should always be used for steam turbines or any other circulatory system where trouble may arise from these causes. Simple physical or chemical tests to determine the suitability of an oil under such conditions are practically unknown except for the common test of mixing oil and water together in a bottle and seeing how quickly they separate. Even though this test might show good results the first time it is tried, it is no reliable indication as to what is going to happen after the oil has been used for many days, weeks or months.

Columbus Coke Separator Recovers Unburned Fuel

As a result of the great need for economy of fuel in Europe, particularly in Germany, a machine was developed in the latter country late in 1920 for separating unburned coke from ashpit waste. It is reported that about 225 installations have been made in Germany, and more recently the machine has been introduced in other European countries. It is now being put on the market in this country by the R. Stutzke Co., Inc., 526 West 18th St., Chicago, Ill., and is being put forward as a measure of fuel economy that is now justified by the increasing need for such economy in this country. It is pointed out that the combustible matter remaining in industrial furnace residues may reach from 20 to 40 per cent or more and that the recovery of this unburned fuel is desirable if it can be done cheaply enough.



MACHINE FOR SEPARATING UNBURNED COKE FROM ASHES

In the inclined tank the coke, being lighter, rises to the surface of the treated water, and the slag, being heavier, sinks to the bottom. The two conveying screws then deliver the coke and slag into separate cars.

The principle of operation of the Columbus coke separator is based on the difference in specific gravity between the unburned coke and the slag contained in

the residue from the furnace. The separation takes place in water that has been raised to 25 or 30 deg. Baumé by the addition of clay, loam or carbide slime, etc.; when this liquid is at the right specific gravity the coke being light, rises to the top, but the slag, being heavy, falls to the bottom.

The whole process, which may be followed through in the illustration, is as follows: The furnace residue is dumped into the revolving drum sieve, from which the fine cinders and ashes sift through into the first dump car. Large cakes of slag (which contain very little coke) pass on through the smaller part of the sieve, out of the end and into a second car. All the rest goes through the coarser screen in the smaller part of the sieve, falling into the end of an inclined tank that is half full of water treated as previously described. There are two conveying worms in this tank, one above the other; the lower one, which extends to the extreme bottom, carries out the slag to a dump car, while the upper one, which goes just below the surface of the water, carries off the coke to another car.

The coke, which is only slightly moistened, can be burned at once; and the cinders and slag, it is said, can be made into an excellent building material in the form of bricks or blocks, after being mixed with about 10 to 12 per cent of cement.

The machine is made in three sizes, yielding from 1½ to 12 cu.yd. per hour, requiring from 1 to 4 hp. It is said that only one unskilled laborer is required for operating even the largest size, and that almost any waste water, fresh or salt, may be used, about 20 to 50 gal. per hour being required. The two smaller sizes are portable and so may be moved from place to place or up to the existing slag heap. Power may be applied either by belt drive or by electric motor.

The Automatic Hydro-Electric Generating Station*

The discussion of the broad features of any hydro-electric development today includes consideration of the advantage and disadvantage of making it entirely automatic. The station may be so large that the cost of manual control is so small relatively as to make it inadvisable to take the slight risk involved and incur the additional expense in the automatic station. The station may include complicated switch gear and require operators of good judgment whose service cannot easily be dispensed with. It may be the new center of distribution of the system, or automatic control may not be a good business policy, having in mind the character of the load.

From a technical standpoint, however, there does not seem to be any limit to the capacity of generating stations for which automatic control equipment can be designed. Manufacturers of equipment—both hydro-electric and electric—recognize the interest in, and appear ready to discuss the details of, any kind of development. Much important work has already been done in connection with automatically controlled synchronous-converter substations.

In medium and small-sized plants there appears to be a great field for automatic equipment. Such plants, when auxiliary to larger systems, can reduce the number of operators to a watchman or eliminate them entirely, depending upon the regular inspection that is necessary even in an automatic plant to anticipate routine trouble, or depending upon the equipment shutting down if in trouble. No difficulties are experienced with the elimination of operators.

With further expansion automatic stations will contribute substantially to the increase of small water powers where the development costs for small amounts of power are relatively low and where the cost of maintaining operators is a serious handicap. With such developments logically go automatic pond-level control devices, such as siphonic spillways and collapsing crests.

Two schemes of control are in use and under discussion: First, the remotely controlled automatic station in which the generator leads run to the main system and the generator is synchronized by hand; and second, the entirely automatic station, where the generator is automatically synchronized in the transmission network.

Manufacturers of hydraulic and electrical equipment report increased demands for automatic stations. The electrical manufacturers have for a long time furnished regularly all the devices making up the automatic features. In connection with remote control in standard stations and full-automatic operation for rotary converter stations, all the electrical devices used have been fully developed. Improvements, however, are still being made. An automatic generating station not previously described is that now being built for the Ford Motor Co. at Flat Rock, Mich. At this plant the head is low and controlled by siphonic spillways. The stream flow is irregular, and the amount of power is low and amounts only to a small part of the total demand on the system of which the hydro plant is a part.

This station will contain two 500-hp. units, each with its own independent governor system. This plant falls in the class of entirely automatic stations, the number of units in operation and the loads being determined by the position of floats in the forebay. The equipment shown consists of automatic switching and protective devices that have long been produced for other applications.

Protective devices may be divided roughly into two classes—those that guard against dangers from outside and those that guard against dangers inside the station. In general, the trips protecting against external dangers automatically come back to running position and allow operation to continue, while those protecting against internal dangers usually require an inspection of the equipment and resetting to allow operation to proceed.

Automatic stations require protective devices to guard against dangerous abnormal conditions. These devices always include protection against overload, overspeed and hot bearings, and may include others, depending on local conditions. In the station at Flat Rock protection is supplied against overload, overspeed, hot bearings, reverse current and low oil pressure in the governor.

The best-known automatic generating station is that at Cedar Rapids, Iowa. There are also a number of small ones in which operation is said to be entirely satisfactory, but none are old enough, perhaps, to supply conclusive evidence of their reliability. Sufficient is known, however, to encourage the planning of relatively large ones, and there is every indication that consideration of an automatic plant is now part of the preliminary discussion of every hydro-electric project.

*Abstract from N. E. L. A., **Hydraulic Power** Committee's Report.

EDITORIALS

Noise Prevention Worth While

"I HAVE got so used to this racket that I don't mind it any longer." Everyone has heard such an expression as that from men working in a din of clattering machinery that drives away those not accustomed to it. It is undoubtedly a fact that in this case, as in others, "familiarity breeds contempt"; a man who is exposed all day long, day after day, to the same noises naturally accustoms himself to them, no matter how distracting they may be at first, and finally reaches a state where he is conscious of no serious inconvenience. A general understanding of this fact, however, has led too easily to an unsound conclusion.

The fact that a man no longer feels discomfort at the noises around him does not mean that they no longer have any effect upon him. The whole matter was gone into thoroughly by Henry J. Spooner, an English mechanical engineer and a member of the International Committee on Industrial Fatigue, in an article in *The Nation's Health*. Prof. Spooner emphasized particularly that the bad effects of noise continue, regardless of whether or not we are conscious of them. He says further, that even in sleep, noises that may not awaken one have a tiring influence. The point is that noises around a plant may have become familiar, but continue to have their devitalizing influence upon all those who have to listen to them.

The average power plant is not a noisy place, and sometimes, in fact, it has almost a library atmosphere. Still there are some plants, and some parts of other plants, where no one but a deaf man would enjoy the work. And in cases where the noise is really great, the workmen are in a fair way to become deaf, since it is well known that trunk makers, riveters, boilermakers, etc., very often lose their hearing. Some noises are unavoidable, but many can be eliminated, and some may be stopped by ridiculously simple means.

The increasing complexity, size and power of modern machinery require from those who handle it the utmost of clear-minded alertness. "Manufacturers and others should realize that noise prevention and reduction is a paying proposition," says Prof. Spooner. To manufacturers it means more production and less spoiled work, but to the man responsible for the power plant it means better insurance against accident, increased contentment among employees and more skillful and economical handling of the apparatus.

Higher Steam Pressures

IN 1915, when high steam pressures and temperatures were being actively discussed, one school of engineers predicted a rapid advance to five hundred pounds and seven hundred and fifty degrees, while their more conservative colleagues were content with two hundred and fifty pounds and six hundred degrees as the most desirable limit.

Viewing the field seven years later, one sees no plant operating at the upper figure, while there are several among those recently constructed whose designers have adhered to the lower. Nevertheless, there has been a general upward trend to the extent that there are now operating in this country seven plants at pressures over three hundred pounds and temperatures over six hundred and fifty degrees. While some of these have been designed for as high as three hundred and fifty pounds pressure, it has not generally been found best to operate them above three hundred and twenty-five. The trend in European practice appears to be along parallel lines.

Troubles with fittings and those resulting from misalignment and unprovided-for expansion, encountered in the early stations using high temperatures, are now fairly well understood and the solutions have been largely worked out. That the character of load has much to do with satisfactory operation is also recognized, especially with regard to the turbine units where rapid variation of load is likely to result in undue expansion and contraction in the low-pressure element.

As a result there appears to be little inclination at present to exceed six hundred and fifty to seven hundred degrees, but engineers are again talking of going to higher steam pressures, in fact one large company is understood to be actually preparing the plans for a station to operate at five hundred pounds.

Steam turbines are most efficient in their low-pressure ends, but the practical limit in vacuum has been reached, hence further attempts to secure economy must be made in the high-pressure element. However, increasing the pressure without increasing the total temperature means less superheat. This, in turn, brings the dew point of the steam farther up in the turbine. As the losses increase with both density and wetness of the steam, one must consider to what extent these will counteract the gain due to increase in pressure. Undoubtedly, provision must be made to remove the moisture or reheat the steam, as is being done by Ferranti in England, but this increases the length of the shaft and tends to increase the liability to expansion troubles with the existing high-temperatures.

It is significant, however, that to date the best plant performances, according to available figures, have been in the plants using more conservative pressures and temperatures.

The Solution of Alternating-Current Problems

POWER factor of alternating-current circuits is taking on such significance in power-plant operation, that it behooves every operating engineer to familiarize himself with its use and its effect on the power system. In a direct-current system the total load in kilowatts is equal to the arithmetical sum of the component loads, and this is also true for an alternating-current system. However, the kilowatt load on an alternating-current system is seldom equal to the kilovolt-ampere load, on which the loading of alternators, transformers and size of conductors must be based.

Unlike the kilowatt load the kilovolt-ampere load on

the system cannot be obtained by taking the arithmetical sum of the component kilovolt-ampere loads. Likewise, the composite power factor of a number of loads making up the total on the system is not the average power factor of the component loads. To calculate these quantities would involve considerable work and a knowledge of trigonometry, but there is at hand a method that can be easily applied by anyone who can lay off and measure distances accurately on a sheet of paper. The article, "Power Fact or Correction by Rearrangement of Motor Drives," by T. H. Arnold in this issue, is an interesting study of how to solve these problems easily.

In the solution of these problems the cosine of the angle of phase difference between the current and voltage, which is the power factor, is always involved. This is something that those who are not familiar with the term, too frequently assume is difficult to understand. However, the table of natural sines and cosines is only a tool, like the steam tables, and to the average operating engineer nothing about the steam table is mysterious even though he may not be able to determine how each set of values has been obtained. The steam tables make a good-sized book, whereas a readable table of natural sines and cosines, sufficient to work out the ordinary power-factor problems, may be contained on an area three inches square. A little study is all that is necessary to show that ordinary alternating-current load problems can be easily and simply solved by the use of a draftsman's scale and table of sines and cosines, by constructing a diagram to scale from which all values can be measured, no calculation being necessary. Furthermore, many of the problems are even more simple than the ordinary problems involving the use of the steam table.

Hoover Furnishes a Moral Yardstick

HONESTY in any business depends upon the ethics and standards of that business which are generally recognized at the time in question. In the coal business there has been no moral yardstick by which to judge, but Secretary Hoover, with his usual appreciation of practical affairs, now offers such a standard in the form of basic fair prices for coal.

In his conference with non-union coal mine operators, the Secretary proposed: First, that the Garfield prices for mine-run fuel, appropriately modified to serve present conditions, be used as a basis of fair prices for coal during the present emergency; secondly, that a committee from each district co-operate to advise him as to modifications of such basic schedules as would be reasonable to meet present conditions; thirdly, that these committees or some successors serve for each district to advise the Secretary of unfair prices charged and to assist him by such means as he might direct in preventing unfair prices. These three proposals have been generally accepted by the coal-mine operators. It now remains to be seen just how they will be applied.

In the past it has been extremely difficult to define profiteering in coal. The mine operator blamed the wholesaler and the wholesaler the operator. Both generally joined in condemnation of the retailer, who in turn divided his criticism between the others in varying measure, according to local circumstances. But basically, the whole trouble has been that nobody knew what emergency costs were and no one had the authority or the willingness to declare what would be a fair price either at the mine or at the market. Now Hoover steps into the breach and supplies that most basic need —a measure by which to judge reasonableness of coal prices.

The coal purchaser certainly can take great comfort in this act of the Secretary of Commerce. It may be that the price of coal will frequently soar above the standard thus set, but it is much less likely to do this when there is a definite standard by which everyone may judge the right and wrong of the situation. In many cases it is not unlikely that a bonus above the fair price will be asked or will be offered, but in every such transaction the purchaser undoubtedly will be making mental note of those who accept such excess in price with the thought that after the emergency is over he certainly does not expect longer to deal with any man who insists upon a higher return than has been judged to be fair and reasonable for the emergency period.

This moral force of public opinion among coal purchasers is of tremendous importance. It will be well worth while for the coal purchaser to recognize it now as well as later. In fact, it may be that widespread recognition of it will do as much as the words of the Secretary himself to prevent runaway markets. It appears that the Secretary counts upon this force and will welcome its use. There need be no hesitancy, therefore, in applying it to the full.

More Coal Dope

THIS time it is the Fuel Emergency Company of Newark, New Jersey, and the product which it is exploiting is known as 3-C Coal Saver. One of our subscribers writes that an analysis of this material, made by their chemist, reported it as a mixture of common salt, calcium permanganate and some carbonaceous matter, the bulk of the material being common salt.

There is no apparent reason why sprinkling this upon the coal, either dry or dissolved in water, as prescribed, should have any other result than that which might be brought about by its psychological effect upon the fireman.

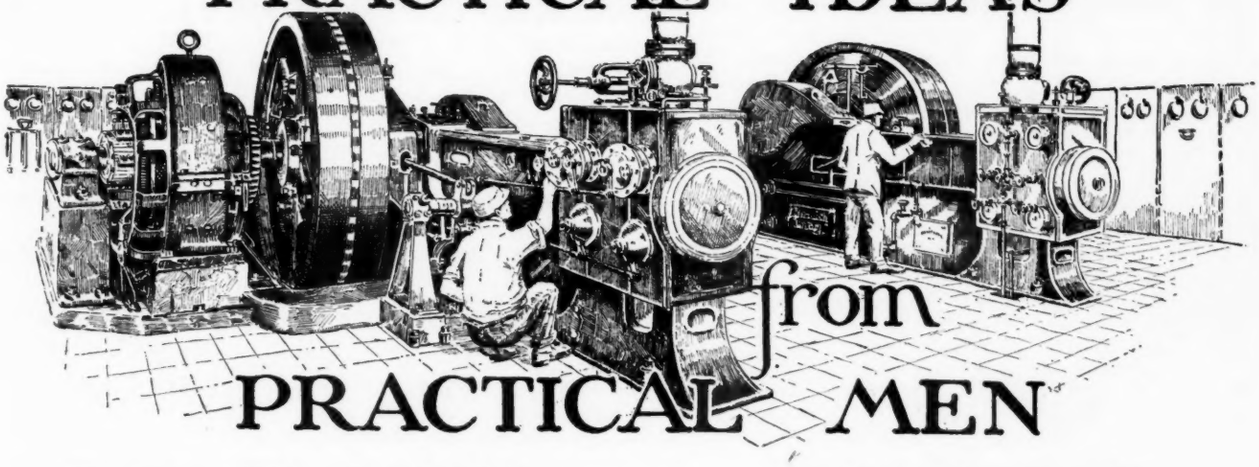
The Government is seeking all available facts about the storage of bituminous coal. It deserves the cooperation of every coal consumer. Turn to the News Section, read what is wanted, and add your own information to that contributed by others. That will help toward eliminating the seasonal buying of coal—and that is at the root of the problem of a constant and reasonably priced coal supply.

The Greeks and the Romans and other notables of ancient times, when they had a hard job to do, went out and captured a few thousand of their enemies and put them to work. Modern applications of power save us the trouble of making our enemies slaves. The kilowatt has taken their place and not only does the work better and more quickly, but is more convenient to have around.

One of the titles in a recent issue of *Power* was "Maintaining Heat Balance at Hell Gate." A subscriber arises to remark that "they will have a devil of a time doing it."

A new kind of engineer is born every little while. The last is the *Bottling and Packing Engineer*. Next?

PRACTICAL IDEAS from PRACTICAL MEN



Restoring a Water Supply Under Difficulties

A water-pumping station in a Southern city originally consisted of two horizontal-tubular boilers, two engine-driven deep-well pumps and one compound duplex high-pressure pump. This plant was later electrified by putting a motor drive on one of the deep-well pumps and changing the other to an air lift and installing a motor-driven air compressor to supply the air. Two centrifugal motor-driven pumps were also installed. This made a nice layout and gave good service for several years.

The electric-light plant, which was owned by the city, was later destroyed by fire, thus cutting off the supply of power to the pumps. This necessitated putting the old steam plant into operation again with the least possible delay. It had been left just as it was shut down; the iron stack had corroded through and most of it had fallen down, the boilers and furnaces were left in a bad condition, arches in the doors were gone, the extension plate in front of the boilers was corroded through, the packing had been left in the pumps, and the rods were badly corroded.

The first thing that was done was to put a force on the boilers, cleaning out of them the scale that had become loose and lodged on the tubes and shells of the boilers, putting in new blowoff pipes, repacking valves, cleaning water columns, placing the grates in shape, repacking the pumps and replacing the valves. After working about sixteen hours, we had the hydrostatic test on the boilers and were ready for steaming, and three hours later were pumping water from the reservoir to the standpipe. We then got one deep-well pump going by disconnecting it from the motor and belting it to the idle engine, but that would furnish only a small amount of water. The next to be put in operation was the air lift. We borrowed a gas tractor and belted it to the air compressor. This did not give sufficient power to run the pressure up high enough, so we removed the suction valve on one end of the compressor and ran it single-acting. This gave us the required pressure, but a great reduction in the flow of the well. We then secured two air-brake pumps from the railroad and put them in multiple with the compressor. In this way we kept the city supplied with plenty of water at all times.

C. D. DISPENNETTE.

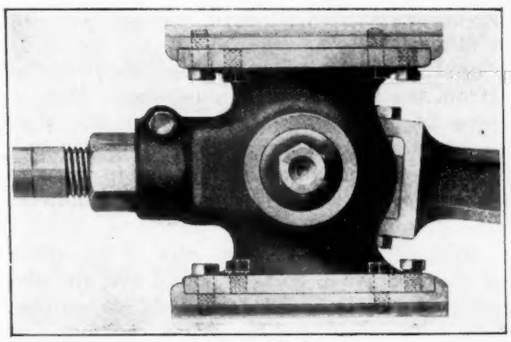
Richmond, Ind.

Providing Adjustment for Crosshead Shoes

In industrial plants a shutdown is occasionally necessary, owing to lack of material or other causes. When this takes place there are usually some repairs to the engines or equipment that can be made.

Wear of moving parts is continually going on and many of these are not provided with means of adjustment. With modern appliances available, such as gas welding, thermit welding and portable power-operated tools, many worn parts, if repaired in time, will be as good as new.

There was originally no way of taking up the play in the crosshead guides of our engine except by removing the shoes and replacing with new ones, or by



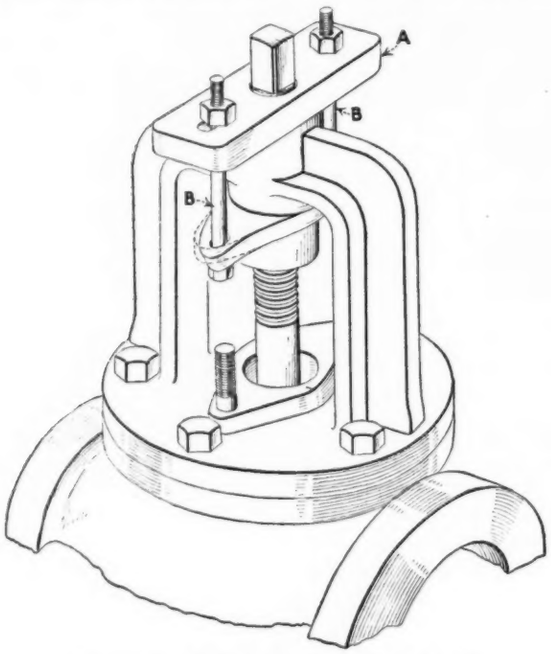
ADJUSTMENT IS MADE BY THE FOUR SHORT CAPSCREWS THEN HOLDING BOLTS ARE TIGHTENED

shimming up. Taking advantage of a shutdown when waiting for material, I removed the shoes and made means of adjustments as shown in the illustration. Both the top and the bottom sides of the crosshead casting were drilled with six holes. Two of these on each side were drilled clearance size for ⅝-in. capscrews, while the other four were drilled tapping size for ⅝-in. capscrews. The screws were arranged so they could be conveniently reached when adjustment was necessary. The shoes can be brought to the correct adjustment and locked without one corner being made too tight by the locking process. The short screws merely tighten on the base of the shoes and after these have been set to the proper adjustment the holding screws are drawn tight.

North Bend, Neb. G. G. MCVICKER.

Straightening Bent Glands on Small Valves

The glands on small valves having outside yokes are frequently bent and in many cases are scrapped instead of being straightened. On glands that have a slot in the end for the studs instead of the drilled hole, this bending is particularly troublesome as the studs



METHOD OF STRAIGHTENING GLAND WITHOUT REMOVING FROM VALVE

for this type are usually hinged and they slide out of the gland when being tightened, while in the gland having the drilled hole, it is difficult to take the gland off the studs.

One method of straightening a gland without removing it from the valve is here illustrated. Make a strap as shown at *A*, slightly larger than the gland and capable of standing a greater bending stress. Drill three holes corresponding to the holes in the gland but $\frac{1}{16}$ to $\frac{1}{8}$ in. larger. Then remove the handwheel and place the plate over the valve stem and on top of the yoke. Release the gland and pull it up until it is against the yoke, then bolt the gland and plate together while in this position with bolts *B* of the same size as the studs. Then pull up evenly on the nuts until the gland is straight, as shown by the dotted lines.

The bolts may have to be bent slightly to get them through the holes; this will depend on whether the gland has slots or holes for the studs.

Struthers, Ohio. J. V. EMERY.

Who Should Supply the Tools?

There is one question on which I have seen no comments in *Power*, and that is the right or otherwise of an engineer to supply the tools for use in the plant where he is employed. I know engineers who own and use valuable tools and instruments, including steam-engine indicators. They know that if these things are destroyed or broken, it will be their own loss. In my opinion the employer should supply the tools.

All will agree that tools are necessary if the work is to be done properly and promptly. They are not necessary for the financial benefit of the engineer, but they are for the owner of the plant. A safe rule to follow, and one that might obviate trouble and misunderstanding later on, is to state their value to the owner, then come to some agreement as to their repair or replacement if they are damaged or lost.

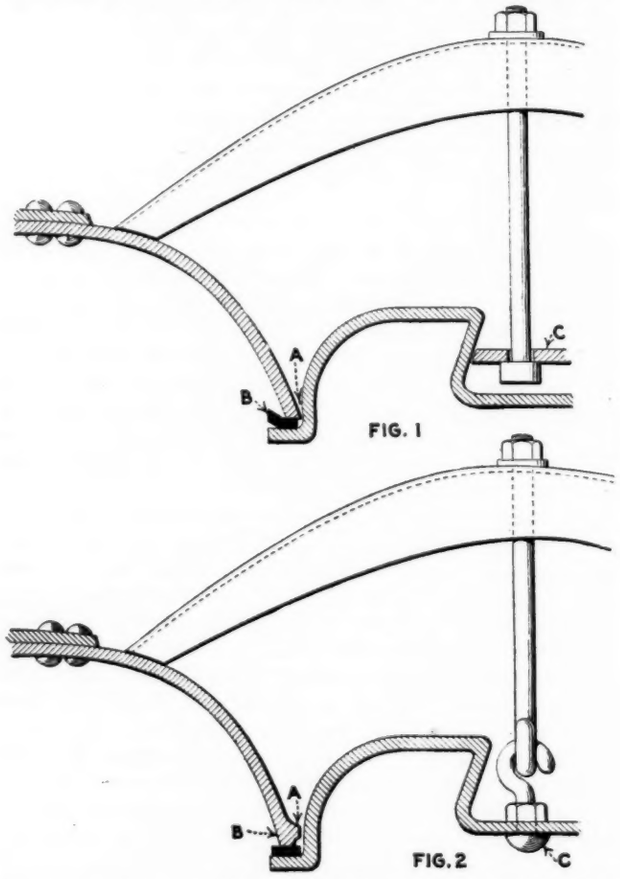
I know of several cases where serious trouble developed over privately owned tools. In one case the employer contemplated having his engineer placed under arrest when he undertook to remove his own set of tools and instruments from the plant.

Toronto, Ont., Canada. JAMES E. NOBLE.

Changing Manhole to Obtain Tight Joint

In the operation of a water filter we experienced considerable trouble in making a tight manhole joint until some changes were made, as shown in the illustration. Fig. 1 shows how the manhole plate fitted close to the flanged head as at *A*. This caused the gasket *B* to be forced outward. The plate *C* was of steel with a slot for the clamping bolt, and when the bolt was tightened up, the plate *C* would spring.

The changes made to overcome the difficulty are shown in Fig. 2. The flanged head *B* was heated by an



FIGS. 1 AND 2. SHOW CLOSE-FITTING MANHOLE PLATE AND CHANGES MADE TO OBTAIN TIGHT JOINT

acetylene torch and hammered in to give more space for the plate. Four lugs *A* equally spaced were welded on to centralize the plate, and the eye-bolt *C* was put through the plate and riveted. After making these changes, the joint gave no further trouble.

Latrobe, Pa. D. R. KOONTZ.

COMMENTS FROM READERS

Penstock Gate Valve Fails and Causes Turbine Wreck

The description of the hydraulic-turbine failure at Niagara Falls, published in *Power*, May 9, brings to mind how a minor trouble is liable to be followed by a serious smash-up, as is shown by the following occurrence in a large hydro-electric power plant in western New York. A section through the 9,000-hp. Francis-type turbine operating under a 256-ft. effective head at 327 r.p.m. and the 66-in. hydraulically operated gate valve for separating the scroll case from the penstock are shown in Figs. 2 and 1 respectively.

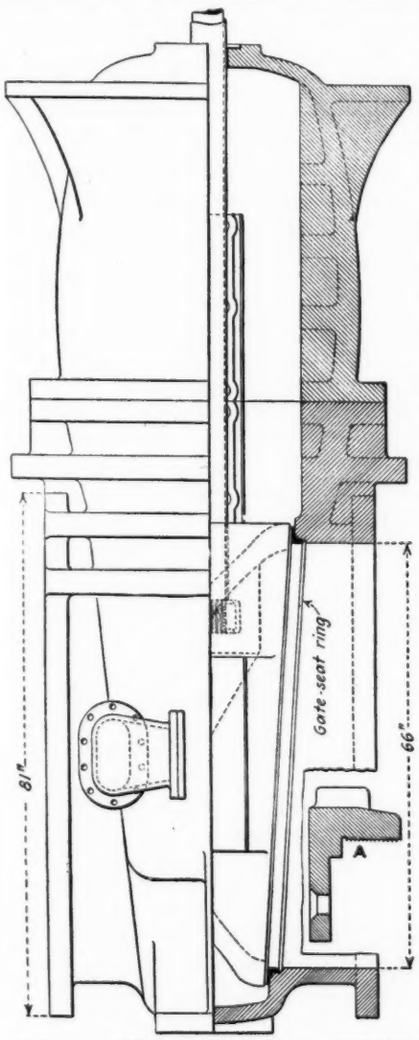


FIG. 1. SECTION THROUGH PENSTOCK GATE VALVE

Presumably by the repeated operation, the bronze gate-seat ring on the downstream side of the valve was broken and parts of it washed through the guide vanes into the runner, which carried the fragments around for some time as the brass marks on the back of the guide vanes indicated. In spite of the missing seat ring the gate shut almost perfectly, thus hiding the facts of the real cause of the trouble until the operating engineer found a piece of the ring in the scroll case.

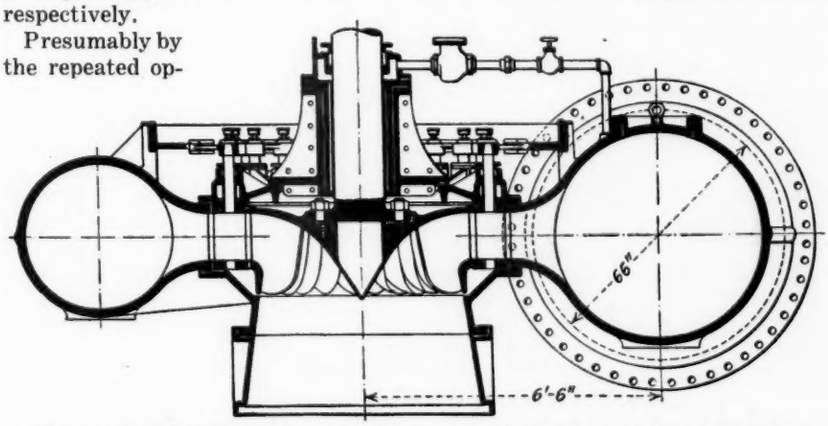


FIG. 2. SECTION THROUGH 9,000-HP. TURBINE WHICH WAS WRECKED BY FAILURE OF GATE-VALVE SEAT

Practically every guide vane (cast steel) was either cracked or at least had the tips bent toward the outside. Two of the guide vanes were broken completely off their shanks and evidently happened to get into the runner in such an effective way that every bucket was broken from the hub and band. All ruptured surfaces showed sound material.

Naturally, no running away of the generator took place since it could operate only as a motor on the line, as the runner could not furnish any power. The only outward sign of trouble was considerable noise. Fortunately, the crown plate and scroll withstood the strain, thus saving the entire plant from being flooded. The scroll case of the turbine is set in concrete, and the pit on top of the scroll case reaches to about the center line on top of the scroll. In other words the scroll case is well secured and the surrounding concrete adds considerably to the strength of this part.

Fig. 1 shows the design of the gate valve and seat ring, which is fastened to the valve body by a thread and is of rather light design with little strength against collapsing, particularly in the upper region. An enlarged section through the seat ring is shown at *A*. It is easy to see that at the upper end of the valve stroke the top edge of this seat has to transmit the friction from the gate to the valve body. For this reason alone the seat ring should be heavier; furthermore, it should be fastened to the valve body by means

of stronger bolts or the metal peened into a dovetailed groove. The breaking out of the seat ring was also experienced on other valves, some of them being installed about twenty years, but without the disastrous result described.

The crown plate, which had been in use since March, 1919, showed serious corrosion of the ribs and other places, as indicated in Fig. 3. Holes occurred near the

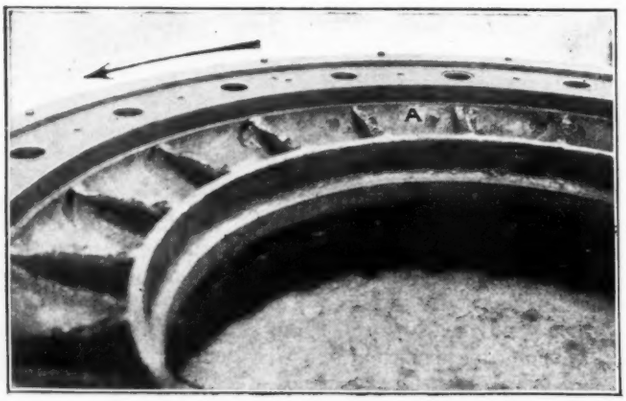


FIG. 3. SHOWS CORROSION ON INSIDE OF CROWN PLATE

ribs which are about 1½ in. in diameter and ⅜ in. deep and have been caused by the leakage water spouting through the clearance between the runner and crown plate. The general direction of this corrosion on the surfaces indicated by *A*, Fig. 3, is at an angle of about 45 deg. to the radius of the crown plate. The concentration of the corrosion at the spots where the holes are is an unusual occurrence, while the eating away of the ribs is easier to understand. The curved arrows indicate the direction of rotation.

Another interesting corrosion occurred on the top of the runner where a piece of lead 8 x 5 x ¾ in., indicated at *A*, Fig. 4, was placed for the purpose of balancing

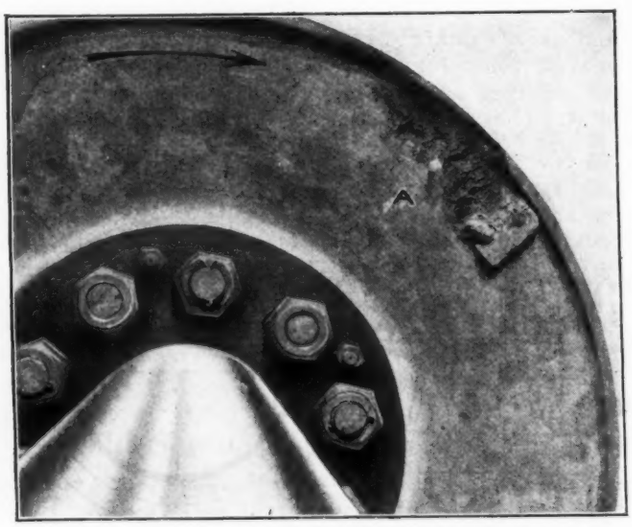


FIG. 4. SHOWS CORROSION ON TOP OF THE RUNNER

the runner. The maximum corroded depth in the casting is ¾ in. It would be a rather far-fetched assumption to charge this corrosion to electrolytic action although metals of different character are in connection.

All these corrosions lead to the conclusion that in spite of the ribs on the under side of the crown plate, the water must still have had a considerable velocity, which probably was augmented to some extent by the square-cornered balance piece. Furthermore, all corrosions are, as usual, in such a place where the revolving water can cause a vacuum. For high heads and high revolutions such a balance piece should therefore be well chamfered at both ends. A better method would be to place the runner eccentrically on the boring mill and turn the proper amount of metal off to give the desired balance. This runner was placed in the machine about February, 1921. Other corrosions which occurred in the runner vanes have been fully described in *Power*, May 24, 1921.

C. VOETSCH,
Hydraulic and Electrical Engineer.
Utica, N. Y.

Making Ammonia Compressors Safe

I have noticed the article in *Power* of May 9, entitled Making Ammonia Compressors Safe, and it seems to me that it should not go unanswered, as it might lead the inexperienced to change compressor valves from one type to another and obtain, not better but worse results.

It is true that with the Linde type of compressor poppet-valve stems do break and accidents occur, but such accidents are not common as is evidenced by the comparatively small number reported during any year.

If high-speed valves could be substituted for the slow-speed valves with the results mentioned by the writer of the article, there would be few slow-speed compressors not changed to the high-speed type, but inquiry will show that few have made the experiment. The reason that poppet valves are still used is not hard to find.

A high-speed type of compressor valve for a given diameter of cylinder opening has a much smaller valve opening than a slow-speed valve of the same diameter. Therefore unless more of the high-speed valves are used, there will be a much reduced valve area, and consequently a reduced volumetric efficiency at any speed, due to the withdrawing. Furthermore, there will be a decrease in capacity at any speed due to the large clearance space contained in all high-speed valves. These two reasons—small valve area for a given diameter valve and greater clearance space in the valve—account for the fact that the high-speed compressors have many more valves than the slow-speed compressors of the same per minute piston displacement. They also have a smaller refrigerating capacity due to the clearance.

Getting down to figures, which are more conclusive than argument, a slow-speed valve of the poppet type 5 in. in diameter over-all would have an effective valve diameter of about 4½ in. The valve can open a distance of at least ¼ its diameter and thus give an area of about 15.9 sq.in; this is the "restricted" or smallest area.

A ring-plate high-speed valve of the same outside diameter (to fit the same cylinder opening) will have a restricted area of only 2.6 sq.in. for a discharge valve and 5.1 sq.in. for a suction valve. The difference is due to the larger lift permissible in the suction valve, and not the supposed fact that the discharge-valve opening should be smaller than the suction.

It will thus be seen that for any given slow-speed compressor cylinder, the valve area possible with the two types is greatly in favor of the poppet valve, and the latter will hold its own.

G. H. CRAWFORD.

Why Do the Piston Rings Break?

Referring to Mr. Hilliary's question in the Feb. 28 issue, as to the cause of piston rings breaking, the writer had a similar trouble some years ago, and the cause was finally traced to overtravel of the piston, due to the piston rod being too long, causing the outer ring to travel nearly half of its width over the edge of the counterbore.

Accurate measurements showed that the inner ring did not travel to the edge of the crank-end counterbore by an equal amount, indicating that it was due to an error in the builder's shop work.

The trouble was overcome by refitting the taper on the rod sufficiently to bring the piston to its proper position, also cutting off the end of the rod, lengthening the key slot in the rod toward the piston and fitting a small block in the other end of the slot for the key to bear against to pull the rod into the crosshead. When this had been done, there was no further trouble with broken rings.

As to the wear in the bottom of the cylinder, I would suggest that he check the alignment of the cylinder with the crosshead guides, as this may be out sufficiently to cause the piston to bear heavily on the bottom of the cylinder.

ALONZO G. COLLINS.

Philadelphia, Pa.

Measuring Corrosiveness of Water

I read with considerable interest the article in the April 25 issue entitled "Portable Kit Measures Corrosiveness of Water," together with a letter of discussion in the May 23 issue, by G. A. de Graff, chemist of the Public Service Electric Co. This article appears to have been of such extended interest that further discussion is merited, particularly of the points raised by Mr. de Graff.

Mr. de Graff is quite right in assuming that the CO_2 content of water is significant in judging its corrosive property. In fact, it is possible to form a general classification of natural waters as corrosive or non-corrosive, accordingly as they contain CO_2 in excess or deficiency from the standpoint of their carbonate content. It does not, however, appear possible to extend this classification to waters that are free from dissolved oxygen, and the classification, or in fact the presence of dissolved carbon dioxide, is significant only in the presence of oxygen.

It is possible that Mr. de Graff has misinterpreted the method, outlined in the article, of collecting the water under suction. Assuredly, it would be extremely bad judgment to use such a vacuum that there would be considerable flashing of the water when it entered the container. It is easy, however, to so regulate the suction that no flashing occurs, and under such conditions it represents a quite workable method of collecting water from a region of vacuum. It is, however, imperative, as Mr. de Graff emphasizes, in handling water that contains very little oxygen, to avoid contamination in the bottle. The simplest way to accomplish this, from the writer's experience, seems either to collect it under a blanket of toluene, displacing the toluene from the surface as the bottle fills, or else to run a considerable volume through the bottle and displace the contents initially contaminated.

The writer has considered the article in question more from the view of an engineer than a chemist and would feel from an extensive field experience in such analysis that the questions of accuracy of measurement raised by Mr. de Graff were perhaps particularly significant to a chemist, but not of significance in affecting the results obtained from an engineering viewpoint, the accuracy of the method being sufficient for what it purports to be: namely, a field method.

The collection of samples at temperatures above the atmospheric boiling point requires some means of cooling below flashing temperatures, as outlined in the article under discussion. The writer has used at various times block pin and copper pipe, and has found no choice between the two as to accuracy.

The suggestion by Mr. de Graff that the starch indicator be added after practically all the iodine is discharged by the titrating solution is excellent and valuable to the user of the titration method.

Jeanette, Pa. J. R. MCDERMET.

Use of Auxiliary Back-Pressure Relief Valve

There has been considerable discussion from time to time among engineers on the use of auxiliary back-pressure relief valves. This is an important question and one that is usually asked engineers on examination for government positions and in public buildings. I have taken these examinations at different times, and the following questions were asked:

1. On an exhaust steam heating system where should the relief valve be placed and where should the back-pressure valve be placed?

2. If you had six pounds of steam on the exhaust and four pounds on the live steam and you had a reducing valve in between, what would be the effect?

3. If you had 80 lb. on the boiler, five pounds in your reducing valves and four pounds on your back-pressure valves, what is wrong?

Referring to the last question, there is a relief valve installed on the branch leading to the atmosphere. This valve must be set for about seven pounds because if it was set at a pressure below the reducing valve, the steam would be blowing through to the atmosphere. I know of one case where this took place and the engineer burned about 300 tons of coal in addition to the limited amount during one winter.

In schools and other public buildings there is usually installed a heating system automatically controlled by thermostats. In order to have uniform temperature it is frequently necessary to have five pounds of steam in the fresh-air ducts when two pounds is required in the various rooms to be heated. In a system like this it is necessary to have separate return lines and separate condensation traps; in addition there must be a reducing valve connected on the lowest pressure branch as well as a relief valve on the branch leading to the atmosphere.

In the case of supplying steam to two buildings, the building where the boilers are installed does not require as high pressure for heating as the building farther away. This necessitates the installation of a reducing valve on the branch leading to the lowest pressure and also a relief valve on the branch to the atmosphere. To the practical engineer the relief valve is named as the back-pressure valve, which is very misleading to him when taking examinations.

New York City. PATRICK MALLOY.

QUESTIONS AND ANSWERS

Forces Due to Pull of Stack Guy

How is computation made of the force tending to pull the anchor of a stack guy wire horizontally toward the stack; also the compressive stress on the stack due to the pull of the guy? J. W.

The force tending to pull the anchor of a guy toward the stack will be equal to the pull in pounds on the guy multiplied by the horizontal distance from the anchor to the stack, divided by the length of the guy.

The force of compression, or vertical downward pull on the stack due to a guy, will be the pull on the guy multiplied by the height above the anchorage where the guy is fastened to the stack, divided by the length of the guy.

Hard Coating in Fire Tubes

What is the cause of fire tubes on return-tubular boilers becoming covered on the fire side with a hard coating? C. N. H.

The coating usually consists of soot that is cemented together by unburned hydrocarbons driven out of the fuel by the heat of the furnace and condensed upon coming in contact with the lower temperature of the tube surfaces; or it may be due to the use of wet fuel or the use of fuel rich in hydrocarbons, or to cleaning the flues with a steam flue cleaner, causing the soot to form a tough hard coating, more especially when the cleaning is done while the boiler is cold and the soot blower is supplied with steam from another boiler.

Size of Steam Header

Should not a steam header have an area equal to the combined areas of the supply pipes of engines or turbines supplied by the header? R. G.

Engine manufacturers usually provide connections suitable for steam velocities of 6,000 to 8,000 ft. per min., and it formerly was the practice to make the cross-sectional area of the steam header equivalent to the combined areas of the feeders, but in modern practice the header is regarded as an equalizer rather than a storage reservoir. In some power-plant designs the pipes leading from the header to the engines are smaller than called for by the steam connections of the engines, and receiver separators are used about three times the volume of the high-pressure cylinder supplied, with the pipes between receiver and engine the full size provided for by the engine builders. By this arrangement a constant flow of steam is obtained, there may be a full supply close to the throttle of the engine, and a cushion is provided near the engine for absorbing the shock caused by the cutoff. With boiler pressures from 125 to 150 lb., a steam velocity of 8,000 ft. per min. is allowed in the header and as much as 9,000 ft. per min. between the header and a receiver. For steam turbines where the flow is continuous and the header is under 50 ft. in length and free from sharp bends, velocities as high as 12,000 ft. per min. are permissible.

Approximate Rule for Suction Head and Suction Lift

What is the rule for height of suction head or suction lift for pumping water of a given temperature? C. W. W.

With a velocity of from 3 to 4 feet per second and a short suction pipe with one bend, water under atmospheric pressure and as warm as about 160 deg. F. will require no suction head, and at 212 deg. F. will require a suction head of 12 ft. For each deg. F. from 160 deg. F. to 212 deg. F. there should be a suction head of $2\frac{3}{4}$ in. At 60 deg. F. the suction lift may be as much as 22 ft. and from 160 deg. F. down to 60 deg. F. there may be a suction lift of about $26\frac{2}{3}$ in. for each deg. F. of the water below 160 deg. F. If the suction pipe is long, the velocity high, or there are many bends or other obstructions, such as a foot valve, strainer or surface condenser, the losses must be estimated and deducted from the available lift computed by the foregoing rule.

Difference Between Pop Valve and Ordinary Relief Valve

What is the difference between a pop safety valve and an ordinary relief valve? J. W.

A pop safety valve has the valve seat and passages for escape of steam so arranged that after the valve opens for the pressure at which it is set to "pop," the pressure under the valve is assisted by the escaping steam to hold the valve open. The valve is thus opened wider than would be the case if acted upon only directly by the pressure within the boiler, and remains open until the pressure in the boiler becomes reduced a few pounds below the pressure at which the valve is set to pop and chattering is prevented. The difference between the pop pressure and the closing pressure is called the "blow-down."

In an ordinary relief valve no provision is made for assistance of the escaping steam to hold the valve open, and therefore the valve closes as soon as the pressure becomes reduced to the pressure for which valve is set to open. Therefore the valve is not held wide open and is likely to chatter unless the pressure in the boiler continues higher than the pressure for which the valve is set to open.

Wear of Valves from Wiredrawing Steam

Will wiredrawing steam injure the tightness of valves? W. L. G.

When steam is discharged from a higher to a lower pressure through a narrow aperture, as through a partly closed valve, the wiredrawing is accompanied by a high velocity of the steam which results in abrasion or cutting of the valve and seat. The effect is to be seen in rapid wear of steam valves and their seats after a small leak is started in a valve used to stop off the discharge of high-pressure steam to the atmosphere, and also in slowly closing engine valves. The wear is to be attributed to friction of the steam against the sides of the opening and is greater with wet than with dry steam.

Dangerous Air Temperature from Throttled Suction Control

With a compressor having a throttling suction control, is there any danger of oil explosions? R. M. D.

The temperature increase during compression of air in the cylinder depends upon the ratio of pressures. With atmospheric suction and 90 lb. gage delivery pressure, the pressure ratio (absolute pressures) is $\frac{105}{15} = 7$. With suction throttled to 8 lb. "vacuum," or 7 lb. absolute, the ratio is $\frac{105}{7}$ or 15. Since the temperatures are as the $\frac{3}{10}$ power of the pressure ratio, or $T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{0.3}$, assuming 60 deg. suction temperature, or 520 absolute, T_2, the final temperature in the first case, is 936 deg. absolute or 476 deg. F. With the 15 to 1 ratio the final temperature would be 710 deg. F. However, this does not allow for the cooling effect during compression which will reduce the calculated temperature by 200 to 300 deg. F. It is obvious that the increased temperature with the throttled suction may ignite any oil in the cylinder or pipe lines, especially in single-compression machines where the discharge pressure is above 70 pounds.

Causes for Excessive Back Pressure

How can the cause be determined when high back pressure of exhaust is shown by indicator diagrams taken from a non-condensing engine? R. L. G.

Excessive back pressure may be due to obstruction to the free escape of the exhaust, or to the leakage of the piston or steam valve of the exhausting end of the cylinder, or leakage of the exhaust valve on the opposite end. By taking a diagram with the indicator connected to the exhaust pipe near the engine, the same or nearly the same indication of back pressure would signify that the exhaust pipe beyond that point was insufficient in size for its length and number of elbows, or that it was improperly dripped or otherwise obstructed, as by a partly closed back-pressure valve or from receiving the exhaust of other engines. But if such diagrams showed a decided drop of pressure below the diagrams taken from the cylinder of the engine, it would be concluded that the exhaust was choked by insufficient opening of the exhaust valves, contracted exhaust passages or excessive leakage of the piston or valves.

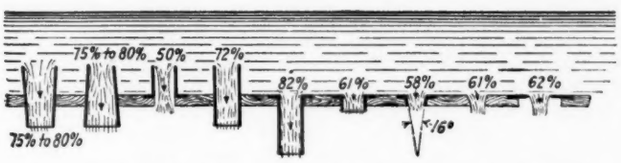
The first of these to receive attention should be the opening of the exhaust valves. With a throttling governor slide-valve engine or exhaust valves operated by a fixed eccentric, the amount of exhaust-valve opening should be actually measured when the engine is turned over by hand or by swinging the eccentric around on the shaft; and when the exhaust is controlled by a shaft governor, the exhaust openings should be determined by blocking the governor to the position of average cutoff and determining the valve openings when the engine is turned over by hand. Leakage can be tested by blocking the engine with the crank far enough ahead of dead center for the steam valve to be open on one end and the exhaust valve open on the other end and observing whether there is escape of steam from the exhaust pipe; or with both of the exhaust valves closed, observing whether there is escape of steam from the indicator cock on the end opposite to the end in which steam is admitted. Leakage that may be due to the piston alone can be seen by removing the cylinder head, blocking the crosshead at any point of the stroke at which it is desired to test the leakage, and admitting steam to the crank end of the cylinder.

Having eliminated exhaust pipe, exhaust valve opening and valve leakage as probable causes for excessive back pressure, the fault must be laid to choked passages for the exhaust in the cylinder casting, due to improper coring or presence of some obstruction.

Water Discharged Through Orifice in Bottom of Tank

What would be the rate of discharge of water into the atmosphere through an orifice 2 in. in diameter in the center of the flat bottom of a tank that is 6 ft. in diameter and in which the water is maintained at a level of 5 ft. above the orifice? G. R.

The theoretical velocity of discharge is given by the formula $v = \sqrt{2gh}$, in which v = velocity in feet per second; g = 32.2, acceleration of gravity; h = head in



PERCENTAGES OF THEORETICAL DISCHARGE WITH DIFFERENT FORMS OF ORIFICES

feet, or $v = 8.03\sqrt{h}$. Substituting 5 ft. for h gives $v = 8.03\sqrt{5} = 8.03 \times 2.236 = 17.95$ ft. per sec.

Hence with an orifice 2 in. in diameter the discharge based on the theoretical velocity would be at the rate of $\frac{(2 \times 2 \times 0.7854)\ (17.95 \times 12) \times 60}{231} = 175.76$ *gal. per min.*

The actual velocity, and therefore the actual rate of discharge, would be less than the theoretical, the percentage depending on the form of the sides of the crifice. The approximate percentages for leading forms of smooth-sided and sharp-edged orifices are shown in the illustration. If the orifice is made with sharp edges, like the one shown at the extreme right, then the actual velocity and discharge would be about 62 per cent of the theoretical, or 0.62 × 175.76 = 108.97 gal. per min.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Experiences in the Operation of Large Turbines*

Review of the Operating Difficulties—Distortion of the Rotor and Housing—Bending of Diaphragm Walls—Bursting and Cracking of Rotor Wheels—Breakage of Turbine Blades—Rusting and Erosion of Turbine Blades—Steam Leakage—Remedies for These Troubles

THERE have been many accidents to large steam turbines during the last few years, which may be attributed principally to oscillation of the wheels. Inasmuch as distortion of rotors and casing may cause rubbing and consequent heating, it is important to start a large turbine with great care in order that the warming-up process may be completed with the minimum temperature stress in the turbine. Otherwise serious rubbing may occur between the moving and stationary blade rings, and of the labyrinth rings on the balancing pistons of reaction turbines. Rubbing may also take place between the labyrinth packing in the casing and diaphragm walls, and the shaft in pressure-stage turbines. If the rubbing is initially due to a slight bend in the shaft, this trouble may become more serious as the developed frictional heat has a tendency to increase the shaft distortion. Unless the turbine is stopped promptly, permanent distortion of the shaft and damage to the rotor wheels will result.

The difficulties cited can be materially lessened by insulating the shaft from the parts that are in danger of close contact and by making provision for a more uniform dissipation of the frictional heat. For instance, one company provides the labyrinth glands with thin rings of large diameter. Should rubbing occur, a large surface would be available for distributing the heat to the steam that surrounds them and that would carry off the excess heat.

Turbine casings are also subject to serious distortion by stiff piping connections and by inadequate outboard bearing support.

Deflection of Diaphragm Walls

The causes for deflection of diaphragm walls may be classified as follows:

a. Pressure difference on the two sides of the walls. The deflection increases with the load. If the clearance is exceeded, rubbing occurs. To prevent wheel deflections from exceeding the safe allowable limit, it is necessary to increase the wheel-disc thickness in a ratio that varies as the fourth root of the wheel diameter. It is, however, extremely difficult to calculate the probable wheel deflection, inasmuch as this is the resultant of two distortions, that of the disc and that of the shroud ring.

b. The diaphragm is usually supported within a machined groove of suitable dovetail or T form with a certain allowance for radial clearance. At no load or while running noncondensing, the temperature surrounding the wheels in the last stages is much higher than for condensing operation. Sudden loss in load or in vacuum may therefore result in heating the diaphragms more rapidly than the increase in temperature can be transmitted to the casing. If the expansion of the diaphragm walls exceeds the clearance, there may be an axial distortion, which will cause rubbing. This condition is aggravated by large diameter wheels.

Bursting and Cracks Developed in Wheel Discs

Trouble from bursting and cracks in wheel discs may be due to any of the following causes: (a) Excessive centrifugal stress; (b) oscillation of wheels; (c) excessive temperature stress; (d) defective material; (e) improper design.

During the last few years many cracks have developed in wheel discs as the result of excessive vibration. Serious consequences have been avoided by careful inspection and reconstruction of the wheels. In other instances loss of life and destruction of the machine have followed. Except in the case of overspeeding, most of the cracking was undoubtedly due to the fatigue of the metal. Such fatigue is caused largely by wheel oscillation, the stresses set up thereby being added to those caused by excessive centrifugal force and uneven temperatures.

Ordinary excessive static stress would not have this effect, since it would merely cause elongation of the steel. But local static stresses caused by rough borings for balancing purposes, sharp corners in the blade, grooves or rough machining of any part of the disc and defects in the steel, tend to promote fatigue, so that every precaution should be taken to avoid them.

Oscillation of Rotor Discs

A combination of the stresses mentioned, when the elastic limit is exceeded, must necessarily result in breakage from fatigue. It is therefore of the utmost importance to design the rotating element in such a manner that a condition of resonance cannot occur. The difficulties in this regard are greatly increased with higher peripheral speeds. If the latter exceed 656 ft. (200 m.) per sec., the required thickness of the disc must be much greater than would be necessary for safety when the centrifugal stresses alone are considered.

Oscillation of the discs can occur in several ways, depending on whether the axes of oscillation lie in concentric circles or are located on diameters of the disc.

The former type of oscillation may be caused by repeated axial thrusts that are transmitted from the shaft, in which they may have been set up by lost motion in the step bearing or the coupling.

The latter type may be due to the uneven distribution of the steam flow across the periphery of the wheel, which in turn may be caused by partial steam admission or even by slight inexactitudes in the blades, particularly near the points where the blades are jointed to the discs and shroud rings.

The axes of oscillation usually remain fixed, but vibrations may cause the axes to shift and to set up dangerous stresses. This condition is usually caused by poor balancing.

All of the foregoing conditions may also exist simultaneously, so that the frequency of oscillation is not dependent upon the shaping of the wheel and character of the metal alone, but likewise depends upon the inner stresses and temperature distribution. This fact renders it exceedingly difficult to calculate the frequency, and it is far safer to be guided by the experience gained with existing machines in designing turbine discs than to base the design on theoretical deductions.

Failure of First Rotor Wheel

In some of the older types of turbines there was a tendency for cracks to develop in the first wheel, to which steam was admitted through several nozzles, under control of the governor, when the peripheral speeds were increased to 708 ft. (216 m.) per sec. Although these wheels were amply strong to withstand the increased centrifugal stress, repeated failures occurred from fatigue.

To overcome this trouble the wheels were made heavier, there being ample room in the casing. Further investigation showed that the introduction of guide vanes in the subsequent moving and stationary blading to divert and expand

*Abstract of an article appearing in the *Zeitschrift des Vereines Deutscher Ingenieure*, Berlin, Dec. 10, 1921.

part of the steam passing through the blades to condenser pressure and to guide the rest to the succeeding wheel, increased the capacity without necessitating the use of special steel in the construction to provide for the heavier stresses that resulted.

Failure of Last Rotor Wheels

One company has experienced considerable trouble with the last rotor wheels in large turbines. The average peripheral speed was 787 ft. (240 m.) per sec. and the maximum reached 984 ft. (300 m.) per sec. It is impossible to reduce this speed without sacrificing the economy. Owing to the great length and weight of the blading, it is doubtful whether reinforcement of the shroud rings would be sufficient to overcome the trouble, principally because of the added stress in the wheel hub, which is already weakened by the key. It would therefore be necessary to increase the size of the hub, which would naturally result in increased length of the machine, or it would require that fewer stages be contained in the same length of casing. In either case the operating efficiency would be impaired.

If, on the other hand, the diameter of the last rotor wheels is reduced, there is a similar loss in efficiency.

Owing to the danger experienced in the past with large wheel diameters, there has been a marked tendency to moderate peripheral speeds in the most recent designs.

Many accidents to turbine wheels have been erroneously attributed to defective material. While no doubt the quality of steel declined during the war, it is certain that this is not the prime cause for the troubles experienced. In spite of the inferior quality of the steel breakage would not have occurred if oscillation had not been present.

Testing wheels by overspeeding cannot be positively relied upon as a means for avoiding future breakage. It should simply be regarded as a precaution that is valid only if no oscillation occurs in actual operation.

The wheel disc should be made as stiff as possible to avoid vibrations in service. One method of testing whether this condition has been met is to support one end of the rotor, before being placed within its own casing, upon the casing of another machine in which the speed may be varied up to that of the rotor. If resonance is present, vibrations in the rotor will point to the fact that similar vibrations are to be expected when the rotor is placed in operation, and which may even be increased in the presence of uneven temperature stress.

Uneven Temperature Change in Large Diameter Wheels

In impulse turbines the rotor wheels are heated or cooled by the steam leaking past the packing of the diaphragm walls. This condition would result in a fairly even temperature distribution through the wheel, but for the fact that the heavier metal at the hub retards a temperature change, whereas this may be quite marked at the thinner periphery. Consequently, in large-diameter wheels the stresses may become quite severe. They are usually not very great in wheels near the exhaust end. The stresses are probably greatest in low-pressure turbines which are subject to greater temperature changes, and because the steam in the last stages, owing to steam leakages is likely to become superheated. It is also possible for the steam temperature to rise suddenly from 104 deg. F. (40 deg. C.) to 203 deg. F. (95 deg. C.) in changing from condensing to non-condensing operation.

In some makes of reaction turbines rotor-wheel discs are employed in the low-pressure end. The space between these discs is filled with steam at rest, so that any load changes will bring about much greater temperature differences between the wheel hub and the rim. The danger of vibration and distortion is therefore greater.

Breakage of Turbine Blades

Turbine-blade breakage may be caused by any of the following troubles: (a) High centrifugal and bending stresses; (b) oscillations; (c) defective materials; (d) defective design and construction.

Most of the trouble appears at the wheel perimeter, where the sharp corners of the T-groove in which the blades are fastened, are subjected to heavy local tensile stress. Oscillations may be caused by synchronous swinging of the rotor wheels, by poor running balance, by defects in the step bearing or coupling or by improper blade design.

Correction of these troubles depends on the causes. If the oscillations originate in the blading, it may be necessary to alter its shape in order to change the time of blade oscillation, or stiffening may be resorted to. In very large turbines the blades must be heavily reinforced.

Rusting and Erosion of Turbine Blades

The blades mounted on the last rotor wheels in large turbines are made of nickel steel, containing up to 5 per cent of nickel. With good feed water free from carbonic acid and oxygen, these blades will remain practically free from rust provided the turbine is kept internally dry during shutdowns. This is accomplished by operating the air pump for some time after the throttle has been closed and to provide the proper means for draining the condensate and for venting the steam leaking past the throttle. The injection of kerosene has also been resorted to.

Erosion due to wet steam can be prevented by draining every stage. All pockets in the casing should be provided with drop points to insure the positive removal of water. Particular care should also be taken in draining the throttle and governor valves.

Lubrication and Bearings

The circumferential shaft speed within turbine bearings has increased from about 59 ft. (18 m.) per sec. to more than double that amount. At the same time bearing pressures have increased from 55 lb. per sq.in. (4 kg. per sq.cm.) to more than 117 lb. per sq.in. (8.5 kg. per sq.cm.). These changes have resulted in comparatively little trouble, though occasionally the babbitt has been melted owing to failure of the oil pump.

In large turbines the lubrication must be planned scientifically. For safety it is best to provide a separate steam or electrically driven pump, to maintain the pressure and the oil flow, if the self-contained pump on the turbine should fail. Telltales should be installed on the oil flow. Often, thermally actuated electric signals are installed in the bearings to give warning of any dangerous rise in temperature. Under no circumstances should the oil temperature be permitted to go beyond 158 deg. F. (70 deg. C.).

Water-cooled bearings have not been entirely successful, principally by reason of clogging of the piping. When necessary, their installation should provide for the most direct path of circulation possible and be without restricted passages.

The partial filtration system is most often resorted to, in which a certain percentage of the dirty oil is continuously removed, filtered and returned to the system. Precautions should be taken to make the separation of water from the oil as complete as possible.

Governing and Steam Leakage

Large turbines must be provided with a stop valve, a main governor valve and an emergency trip safety valve, which can be tripped by hand as well as automatically by overspeeding of the turbine. The arrangement must be such that the main governor valve closes when the emergency valve is tripped. The emergency valve should be tested each time that the turbine is shut down.

Stuffing-box leakage is detrimental in several ways. This steam condenses on the turbine base and heats the oil to a temperature at which it loses its maximum lubricating value. The accumulated moisture gathers dirt and promotes rusting.

Joint leakage can be overcome by proper construction, a feature which deserves close attention due to the use of higher steam pressures.

Care in Turbine Operation

a. Warming up should be carefully done, in an effort to approach the running temperature of the turbine slowly. This means that the high-pressure end must be warmed first, while the low-pressure end is allowed to remain as cool

as possible. The latter end should be warmed only when the vacuum has reached 12 in. (30 cm.). The rotor should not be turned over until the vacuum has reached 15 in., after which enough steam may be admitted to run the turbine at about 10 per cent of the normal speed. After 15 minutes' operation the speed may be increased to full in increments of 10 per cent per minute. This will produce a condition wherein the low-pressure end is at a higher temperature than when running under load. Load should then be added at a rate of not more than 10 per cent of the normal per minute.

The foregoing is subject to variation depending on type and capacity of the turbine.

b. Vibration on Starting—If this occurs, the speed of the turbine should be reduced and the machine permitted to operate at the lower speed for several minutes before being carefully increased again.

c. Vibration in Operation—The turbine should be kept in good condition. Should vibration or oscillation occur in service from an unknown cause, the machine must be shut down, opened and carefully examined.

d. Lubrication—The spare oil pump should always be in running condition and ready for service in case of failure or accident to the main oil pump.

e. The emergency trip valve should be tested each time that the turbine is shut down.

f. All drain connections must be kept clear of obstructions.

Refrigerating Engineers Meet at Detroit

THE American Society of Refrigerating Engineers held its ninth Western meeting at Hotel Statler, Detroit, May 24-26. A well-balanced and instructive program had been prepared, the attendance was fair and the meeting as a whole a decided success. The opening session on Wednesday morning was taken up largely by routine business, a discussion of the Mechanical Refrigeration Code, with particular reference to safety appliances for CO_2 machines and the presidential address by Harry Sloan, of Milwaukee.

The president referred to the favorable position of the refrigerating field during the present period of readjustment due to the limited expansion that had taken place. Reference was made to the work of the National Research Council and to the representation of the Society in this body. At present the subject of heat transfer was under consideration. Though handicapped for lack of funds the committee appointed to co-operate with the American Society of Mechanical Engineers to establish codes for welding non-fired pressure vessels, had worked diligently. The result of the vote by the membership favorable to joining the Federated American Engineering Societies was announced. The code on municipal and state regulations for refrigerating plants and refrigerants had been completed and would be ready for adoption at the annual meeting in December. Copies had been sent to all members, and criticism from a viewpoint broader than the refrigerating field was requested. It was most essential that the safety code be accepted by the American Engineering Standards Committee, of which the Society was not a member, but through its committee had representation. An event of first magnitude to the refrigerating field was the completion by the Bureau of Standards of the Mollier chart giving the properties of saturated and superheated ammonia more accurately than any preceding data.

Wednesday Afternoon Session

Papers delivered at the Wednesday afternoon session dealt with "Methods of Freezing Raw Water," by Halbert P. Hill; "Thermodynamics of Ammonia Compression," by W. H. Motz, and "Cooling of Theaters and Public Buildings," by Fred Wittenmeier. Mr. Hill discussed a number of methods of freezing raw water that had been tried out but had failed commercially. He told why and pointed out the fallacies in each case. Particular reference was made to a method designed to shorten the time of freezing in the cans by reducing the temperature of the water below 32 deg. and taking out the latent heat of freezing in a separate vessel. Also the resistance offered to the exchange of heat by the layer of ice as it formed in the usual method of freezing was to be eliminated. Water temperatures as low as 27 deg. had been attained, but the slightest agitation caused ice-crystal formation in that part of the equipment where it was not desired.

Fundamental principles underlying the relationship of work, power, heat, temperature and pressure of ammonia during the cycle of operation in the compressor cylinders, with a view toward determining the best operating conditions, points of maximum efficiency, etc., were analyzed in the paper by Mr. Motz. A number of useful charts were incorporated giving the mean effective pressures for various suction and discharge pressures; values of volumetric efficiency, due to clearance, for various percentages of clearance and for different suction pressures, and other data.

The final paper considered the cooling of buildings by means of the CO_2 direct-expansion system. The arrangement of the system was discussed, the capacities required and the power per ton of refrigeration. Reference was made to installations at the Congress and Blackstone hotels in Chicago.

Thursday Morning Session

"Calcium Chloride vs. Mixed Chloride Brines" was the subject of a short discussion introducing the Thursday morning session. The consensus of opinion favored the use of calcium chloride brine, which permitted temperatures down to 12 deg. F. Temperatures of 6 deg. had been reached with magnesium chloride, but commercial mixture did not seem to reach this low freezing point. An ammonia leak would precipitate magnesium chloride, but did not affect calcium chloride. The mixtures were about 10 per cent lower in chloride than the straight calcium chloride, percentages of 63 to 65 as compared to 73 to 75 representing the difference. In addition, the combination of different salts favored electrolytic action. The precipitation from the mixed chlorides often caused enough trouble and expense to more than offset the lower first cost.

A most valuable and comprehensive paper on "Compound Ammonia Compression" was presented by G. A. Horne. All data were obtained from an elaborate series of tests in which every precaution was used to insure accuracy. Tests were conducted at pressure ratios of approximately 3, 6, 8, 10, 13 and 16. All tests except those at the two lower ratios were conducted with a back pressure within one pound of atmosphere and varying discharge pressures. Other conditions were maintained nearly alike in the different tests, so that a horsepower curve could be constructed which would fairly represent the entire range of pressure ratios covered under the prevailing conditions. From the data obtained empirical formulas were deduced for determining the correct ratio of cylinder displacement volumes for two-stage compressors, with and without intermediate liquid cooling. The horsepower curves show actual and theoretical values for simple and compound compression.

Included in the data were test results comparing the effect of liquid forecooling and liquid intercooling. The reduction in horsepower per ton effected varied with the pressure ratios employed. At the lower ratios used in ice making, the gain was so small that the elimination of the intermediate liquid cooler was recommended. Relative to the practice sometimes employed of cooling the low-pressure discharge gas partly or entirely with liquid ammonia, the author showed in two comparative cases a saving of less than 2 per cent in power with a corresponding reduction in available tonnage of over 5 per cent, as the ammonia used in this way was not available for actual refrigerating capacity. Another point established was that equal work

in the cylinders gave the lowest horsepower per ton of refrigeration.

Throughout the tests use had been made freely of the venturi-type meter for measuring the ammonia liquid, and the author had been convinced of its value for determining the capacity of the compressor. It was his conclusion that compound compression was a distinct step in advance, particularly for low temperatures, and even with the higher temperatures used in ice making, offered an advantage.

Mollier Chart for Ammonia

In the absence of E. F. Mueller, C. S. Cragoe, of the National Bureau of Standards, presented the Mollier Chart for ammonia previously referred to. Experimental work to establish correct values had been started eight years ago, but had been delayed by the war. The task of getting up equipment to measure the specific heat of superheated ammonia vapor at constant pressures had been difficult, but data of almost absolute accuracy had been obtained and were available in preliminary form in the chart. Full details and the data in tabular form would be ready about the time of the annual meeting in December.

In the next paper H. D. Edwards gave some of the properties and a summary of available information on hydro-carbon refrigerants known as butane, propane and ethane, comparing them with the refrigerants now most commonly used.

"Fish Freezing," by Paul W. Petersen, was the first paper Friday morning. The author discussed the various methods of freezing with salt and ice, by air in a sharp freezer, direct contact with brine and brine freezing with the fish in cans. It appeared that the quicker the freezing the less the damage to the fiber and the more nearly the fish would remain in their original condition.

An instructive paper on the "Present Status of the Synchronous Motor for Direct Connection to Compressors," a subject arousing a great deal of attention at the present time, was delivered by Theodore Schou. The author discussed, primarily, proper flywheel effect and starting equipment. He proposed a limit of 3½ electrical degrees, plus or minus, as the angular deviation from crank effort that should be tolerated. The electrical disturbances would then increase the deviation to approximately 6 degrees.

As low-speed synchronous motors were designed to give sufficient starting torque as well as high pull-in torque and required low starting current as compared to high-speed motors, it was proposed that they be started directly on full line voltage. Details were given of a full automatic control unit that had been designed by the author.

In the discussion of the paper reference was made to the wide variation between builders as to what constituted proper flywheel effect, the variation being as much as 80 per cent. Standardization was highly desirable, but limits should be placed on current pulsation rather than deviation in electrical degrees, the latter being a question for the builder to take care of. After installation an oscillograph could be used to check results. To inaugurate a thorough study of the question and to insure more uniform practice, a motion was carried authorizing the president to appoint a committee to confer with the American Institute of Electrical Engineers and the Power Club.

Measuring Instruments in Refrigerating Plants

One of the topics offered in the program for discussion was "Measuring Instruments in Refrigerating Plants." The time available was taken largely by L. S. Morse, of the York Manufacturing Co., who reviewed the testing work of the company, dwelling in particular on the new testing plant, its equipment and the difficulties they had had in securing instruments sufficiently reliable for accurate testing. With orifice and venturi type meters for measuring ammonia, brine and water, variations from 2 to 15 per cent had been experienced. Chemical thermometers for low-temperature work did not check within 0.6 deg. at different points of the range. Electrical thermometers for accurate results such as required for brine ranges were particularly unreliable. With a variation in temperature they showed a lag of 2 deg. when compared with a mercurial thermometer. In short, meters of any kind could not be relied upon when absolutely accurate results were desired. Actual weighing or measuring wherever possible was to be preferred.

The entertainment features consisted of the usual banquet Thursday evening and automobile inspection tours, the River Rouge and the Highland Park plants of the Ford Motor Co. being the centers of attraction.

Wide Range of Subjects Covered at Federation Council's Meeting

The Executive Board of American Engineering Council, representing the Federated American Engineering Societies, concluded a two-day meeting at Pittsburgh on May 27 after having discussed an unusually wide range of topics. The meeting brought forth animated discussion as to the possibilities of service before the Federation, and satisfaction was expressed with the results. The part of host was played by the Engineering Society of Western Pennsylvania, in whose headquarters in the William Penn Hotel the sessions were held.

The meeting was presided over by President Mortimer E. Cooley, who gave an account of his recent tours throughout the country, and reported increasing interest in the work of the Federation.

Perhaps the most important action of the Board was the adoption of the report of the Committee on Registration of Engineers, of which Col. Arthur S. Dwight, of New York, is chairman. The committee agreed that mining engineers ought not to be licensed, saying "the reasons which justify the registration of engineers responsible for public works do not apply to mining engineers." The committee was unable to reach an agreement as to whether or not engineers in other branches should be licensed, but recommended that the Federation should take the stand "that such legislation may properly cover engineers and architects charged with responsibility for public works." It was agreed that registration laws should provide reciprocity between the states, and the report further stated that such laws "should provide for classified registration, that the certificate may indicate those branches of engineering in which the registrant is qualified to practice."

The employment service that is now maintained by the Council in New York City, and with which volunteer committees in a dozen cities are co-operating, will, it was decided, be turned over to the administration and control of the four founder societies. The Employment Committee of the Council, however, of which Edwin S. Carman, past president of the American Society of Mechanical Engineers, is chairman, will go on with its investigation of the employment question.

Considerable attention was given to the Federal Water Power situation, which, according to the committee, is in so critical a condition that immediate constructive action by the Government to conserve the public interests is necessary. The Federal Water Power Commission's lack of effective and permanent personnel was considered serious, and a resolution was adopted directing the Council's Committee on Water Power, of which Col. J. H. Pinney, of Washington, is chairman, to place before President Harding the facts of the situation with regard to water-power development in this country.

Independent sponsorship of the movement for the establishment of a national department of public works, in case President Harding does not succeed in his efforts toward the same end, was agreed upon by the adoption of the report of the Committee on Public Affairs, which is headed by J. Parke Channing, of New York. The board pledged its support to President Harding in the meantime in carrying out his plan of government reorganization.

A progress report was made by the Council's Committee on Work Periods in Continuous Industries, which has for over a year been making an extensive survey of the two-shift day in American industries. A study is being made of the relative merits of three eight-hour shifts as opposed to two twelve-hour shifts. The Committee on Procedure was directed to put the mass of material obtained into shape for publication and to report at the next meeting.

Improved business conditions are shown by the employment situation throughout the country among engineers, according to a report from executive secretary L. W. Wallace. Other speeches were made by President Cooley, J. Parke Channing and James T. Brady, publicity director. Other subjects that were discussed included jurisdictional awards in labor disputes, patent legislation, an international engineering congress, aërial research, engineering representation on the Civil Service Commission, foreign relations, waste in agriculture, flood control, more intensive engineering organization, government contracts, the steel industry, and the question of affiliation of American engineers with the engineering bodies of other nations.

The Board adjourned to meet in Boston in the fall on a date to be fixed later.

One Way Out of Coal Troubles

After having made a close study of the coal industry for the National Bureau of Information and Education, J. A. H. Hopkins has made a report containing definite recommendations as to what ought to be done to eliminate the present evils of coal production and distribution. The meat of the report is in the following paragraphs:

The operators are entitled to a fair return upon the capital actually invested; compensation for any legitimate financial risks incurred; and a fair percentage of profit over and above the cost price of the coal when delivered to the consumer.

The operatives are entitled to a living wage; proper living conditions; and reasonable protection against the dangers incidental to their occupation.

The consuming public has a right to demand that coal shall be mined and delivered at its doors at a maximum of efficiency and at the lowest price consistent with the rights of those who mine and deliver it.

In connection with these demands it is also necessary to recognize the obligations and responsibilities which they place on the shoulders of operators, operatives and the public. It is equally necessary to bear in mind that the facts absolutely essential to an intelligent enforcement of these demands must be ascertained and established by those to whom these facts are known. For instance, at the present time neither the public nor the Government knows whether the industry is fairly capitalized. They do not know the exact cost of mining and delivering coal; they do not know what percentage of profit the coal industry is yielding.

It is fair to assume that the operators, by opening their books, can either answer these questions or furnish the information from which they may be answered, and it is of prime importance that they be compelled to do so. The Federal Trade Commission started to ascertain these facts, but the operators obtained an injunction stopping the investigation upon the ground that mining coal was "not an inter-state commerce." If this decision stands, the Government should immediately take steps to appoint a commission specifically vested with the authority to compel the operators and the operatives to furnish all the necessary information and to compel them to submit all their records to a searching investigation. The commission appointed for this purpose should also be charged with the duty of utilizing the information obtained to accomplish the following purposes:

1. To standardize the mines on the basis of their productive capacity, and if necessary to order the closing down of all mines which by reason of their natural limitations fall below this standard;

2. To standardize the cost of living for mine workers and the living conditions which must be supplied to them in order to surround them with reasonable comforts, recognizing in so doing the psychological effect of such surroundings in respect to their efficiency;

3. To standardize a basis of arriving at the overhead costs of producing the coal and delivering it at the door of the consumer, recognizing in this compilation that the standardized cost of living for the miners must be the first and irreducible item of expense;

4. To translate this latter item into a national wage scale, taking into consideration the uncertainty of employment and offsetting this so far as possible by a monthly instead of a daily basis of employment;

5. To ascertain and provide, through Federal legislation, for the charging of a fair percentage of profit (and no more) upon the actual cost price of the coal, including freight charges to destination, this percentage to include all costs of distribution from the point of delivery by the railroad or water carrier to the actual door or the factory of the consumer;

6. To consider the recommending of proper legislation, including, if necessary, a seasonal grading of the freight rates that will eliminate the irregular weekly production, avoid the consequent car shortages and establish, so far as possible, a continuity of employment.

Western Engineers Discuss Power for Coal Mines

At a joint meeting of the Western Society of Engineers with the Chicago Sections of the American Institute of Electrical Engineers and the American Institute of Mining and Metallurgical Engineers, held on Monday evening, May 22, the various angles of central station power for coal mines were discussed by three speakers.

J. Paul Clayton, vice-president of the Central Illinois Public Service Co., showed that the central station had made great headway in the mining field, primarily because of the low load factor in most of the plants (caused in part by the seasonal operation of the mine), poor economy obtained in most of the plants, especially for the smaller mines, and the scarcity of good condenser water.

The advantages of purchased power were listed by the speaker as follows: A reduction of initial investment in power generating equipment, greater flexibility for increasing capacity to almost any extent on short notice, independence from a water supply, reliability of power supply during strikes (to prevent flooding when the labor of the plants is withdrawn), reduction of time required during the year for repairs to power apparatus, and a lower power cost per ton of coal mined.

As to the reasons why power could be sold cheaper to coal mines than the mines could generate it, Mr. Clayton stated that a great concentration of production in large stations results in high economy in the use of fuel. The greater diversity of load, he said, results in a higher load factor for the central-station system.

W. C. Adams, of the Allen & Garcia Co., took a neutral position, in which he held that each case is a problem of its own and must be settled on its own merits; there are mines that could operate their own plant to advantage and others—particularly the smaller mines—that should purchase current. In drawing a comparison, he said, it is important that all charges entering into the cost in either case should be included; reliable figures on both sides, with all charges included, are essential for a fair comparison. For example, in a private plant the charge for fuel should be the market price of the coal, and all fixed charges entering into the cost of generation, no matter how remote, should be included.

In the final paper of the evening J. C. Damon, of the West Penn Power Co., treated the subject from the central-station standpoint as existing in his section of the country, where enormous progress has been made by the central station in the coal-mining field. He said that two of the larger power companies of the Pittsburgh district were supplying over 400 coal mines with about 155,000,000 kw.-hr. per year, but admitted that there are still many well-managed and well-operated mines of all sizes operating their own plants to advantage.

In enumerating the advantages of service over private generation, the speaker insisted that to the operating costs of the private plant all fixed charges having any bearing on the cost of generation should be added. Too often, he pointed out, the direct outlay in either case is all that is considered. As to reliability, which is often of more importance than operating cost, advantage was given to the central station, since interruptions may be more numerous, but are of shorter duration.

In conclusion, the speaker presented some interesting statistics classifying the consumers of the company he represented, which showed that the power load was 88 per cent of the total output and brought in 72 per cent of the revenue. Of this power load, 51 per cent was in the coal-mining field.

News in the Field of Power

Fair Coal Prices Fixed at Washington Conference

Maximum fair spot prices for coal in five districts had been decided upon and announced, at this writing, as the result of Secretary Hoover's conference with the coal operators at Washington May 31.

In the Harlan and Hazard fields of Kentucky, the Southern Appalachian fields of Tennessee and Kentucky, and the Pocohontas, New River, Tug River, and Winding Gulf fields (smokeless coal district of West Virginia) the maximum price was fixed at $3.50 a ton; while in the Alabama district it was set at $2.20 to $2.60 a ton, 25 cents below the final "Garfield Scale."

It was explained that a lower price was set in the Alabama district because production costs in that field have been reduced. Mines in that locality are not running at capacity and Mr. Hoover advised consumers in Southeastern States, together with those having access to water-borne coal in the South Atlantic and Gulf seaboards and those as far inland as St. Louis and Kansas City, to take advantage of the opportunity to get coal from that district.

It should be remembered that these are maximum mine prices and that there is nothing to prevent the selling of coal at lower prices. On the other hand, Mr. Hoover has nothing to prevent their being exceeded except "moral suasion," since there is no law by which he can enforce them.

Representative Ten Eyck Warns Against St. Lawrence Project

An extended speech in opposition to the proposed Great Lakes-St. Lawrence waterway project was recently made in the House by Representative Ten Eyck, of Albany, N. Y. He advocated the purchase of territory in Canada lying east and south of the line of the present center of the channel of the St. Lawrence River from its mouth to its source, and of rights to develop and utilize half of the water power from the river. He pointed out that under the proposed plan the waterway would be in foreign territory, and urged that this was inadvisable, as it would enable a foreign country having such jurisdiction to deny equal rights to this country.

Mr. Ten Eyck declared that the proposed waterway, while allowing electrification of railroads within a thousand miles of its development, would cause the scrapping of steam engines and steam plants, cut down the production of coal mines and change the sources of power of related industries. The cost of this change from steam to electrical development and the scrapping of steam-power plants would, he believes, cost the public more than the harnessing of the St. Lawrence River. He thought these changes should be brought about gradually in order to do the least harm.

He pointed out that title to all water-power rights south of the international line belongs to New York, and that the Government cannot develop them except for commerce. The St. Lawrence, he said, is more than a 75 per cent power project. He said further that the deepening of channels, rebuilding of harbors and installation of proper machinery for handling freight for ocean-going vessels on the Great Lakes will cost over $100,000,000, in addition to expenditures in the St. Lawrence project proper.

In opposing joint ownership of the canal by this country and Canada, Mr. Ten Eyck pointed out that during the war Canada cut off the supply of electrical power to some of our manufacturing plants under the plea that she needed it for war purposes, and that the same conditions are likely to occur under the proposed waterway.

Hoover Boosts Superpower

The time is ripe for a great national program of superpower development. The stretch of economic transmission distances has brought power development from the mouth of our coal mines, and upon our streams, to within the economic reach of our industrial centers and our farms. Few appreciate the industrial revolution that lies pregnant in the 350 miles of transmission that is now well within economic possibilities. Nor does any engineer despair that we shall yet develop transmission to the thousand-mile radius.

—From Mr. Hoover's address by telephone from Washington to the N. E. L. A. convention at Atlantic City.

Electricity for French Farmers

The success that has resulted from the operation of farmers' co-operative associations in France has led an association of farmers and peasants in the region of Saint Martin de Londres to undertake the construction of a hydro-electric project in the gorge of the Hérault. Thirty-seven rural districts will obtain their light and power from this station. The development will cost three million francs. The cornerstone of the dam was laid recently by M. Henri Cheron, Minister of Agriculture.

A report from Consul General A. E. Ingram at Havre attaches significance to this project, as it is the first actual effort to electrify rural areas in France.

New York State Opposes Federal Power Commission

A motion for permission to file an original bill against the Federal Power Commission has been submitted to the United States Supreme Court by Attorney General Newton on behalf of the State of New York, contending that the Federal water power law is inapplicable to waters in that state on the ground that the Government's jurisdiction extends to such waters only in so far as navigation is concerned.

It is contended that the Federal water power act is invalid, unconstitutional and not enforceable in New York, and that the Federal Power Commission should not have authority to regulate and control rates for the sale of electric power developed under water-power projects when such sales are intrastate and not related to navigation. The state also seeks to prevent the Federal Power Commission from designating what licenses shall be granted for water-power development.

National District Heating Program Announced

An interesting program has just been announced for the coming annual meeting of the National District Heating Association at the Breakers Hotel, Cedar Point, Ohio, June 20-23.

On June 20, Tuesday, after reports of committees and other businesses affairs have been completed in the morning, an afternoon technical session will be held at which reports will be received from the Heating Research Committee, the Heat Utilization Committee, and the Nominating Committee, and a paper will be given by J. C. Butler, of the Illinois Maintenance Co., on "Value and Method of Steam Generating Plant Interconnection."

On Wednesday morning the Station Operating Committee and the Operating Statistics Committee will give their reports, and A. C. Fieldner, of the Bureau of Mines, will give a paper on "Fusibility of Ash." The afternoon will be given over to golf, games, sports, etc.

At the Thursday morning session the report of the Meter and Accessories Committee will be given, an address will be made by F. Paul Anderson, Director of Research for the American Society of Heating and Ventilating Engineers, and James A. Donnelly, of New York, will give a paper on "Temperature Control by Fractional Distribution of Steam in Large One and Two-Pipe Systems."

In the afternoon reports will be received from the Underground Construction Committee, the Rates Committee, The Public Policy Committee and the Rules and Regulations Committee.

Officers will be elected and amendments to the constitution considered.

At the concluding session on Friday morning there will be a "Round Table" on "New Appliances and New Methods," and Nelson S. Thompson, of Washington, D. C., will give a paper on "Central Station Service in Federal Buildings Under Control of Treasury Department."

All morning sessions will begin at 9:30 and the afternoon sessions at 2:30. The secretary of the Society is D. L. Gaskill, Greenville, Ohio.

Hydro Plant on Spanish-Portuguese Frontier

Utilization of the power resources on the rivers forming the boundary between Spain and Portugal has been brought nearer, it is believed, by the opening and operation of the plant of the Compania Electro del Lima, at the falls of the Lima River on the Spanish-Portuguese frontier. The development of the important power resources of the Douro River, an international stream, has been prevented by the inability of the two countries to agree on the distribution of the power.

The success which apparently is accompanying the Lima development is expected to pave the way for more important developments on the Douro. The electrical equipment for the Lima project was purchased from the General Electric Co. The turbines are of Swiss manufacture.

The information in the foregoing comes from the American Consul at Oporto, in a report to the Department of Commerce.

Two More Power Applications

The Clarion River Power Co., with offices at 14 Wall St., New York City, has applied for a Federal license to cover a power project on the Clarion River in Clarion County, Ia. The primary power available is estimated to be 12,000 hp. It is the intention of the company to install equipment sufficient to generate 30,000 horsepower.

The Northern States Power Co., with offices at 208 South La Salle St., Chicago, has applied for a preliminary permit to cover the construction of a reservoir and three dams on the St. Croix River. The plan is to locate one of the dams on the site of the existing Nevers dam. Another dam is to be constructed below Kettle Falls and a third dam above the falls. This project has been separated from an application made Sept. 20 by the Northern States Power Co. The primary power available at Kettle Falls is estimated at 11,000 hp. The company intends to install 27,000 horsepower.

A Remedy for Coal Troubles?

Believing that basic readjustments in the coal industry will grow out of the strike, George S. Rice, chief mining engineer of the Bureau of Mines, has brought to the attention of the administration officials dealing with the strike situation the plan which he has advocated for several years looking to temporary restraints in the opening of new coal mines. He believes that intermittency in the coal industry will continue until the development of the industry is brought into step with the country's coal requirements. Mr. Rice expresses himself on the subject as follows:

I propose a temporary restraint in the opening of new bituminous or subbituminous mines which intend to enter into interstate business, by withholding track connections and car services until the prospective operators demonstrate in open court proceedings to the Interstate Commerce Commission that there is need of additional coal production for the fuel needs in the districts served by the railroad or railroads which could provide connections for the prospective mine.

Unrealized War Profits on Coal—Lost Forever

In opinions by Justice Holmes, the United States Supreme Court has decided against the Morrisdale and Pine Hill coal companies in their cases from the Court of Claims contending for prices in excess of those fixed by the Fuel Administration during the war.

The Morrisdale company had asked for recovery from the Government of the difference in the price per ton for coal diverted by the Fuel Administration and coal previously sold under contract at higher prices. The Pine Hill company asked for the difference between the maximum price per ton fixed by the Lever law and the price it could have received had the government not interfered.

The Court of Claims rejected the claims of the fuel companies on the ground that the Lever law fixing prices did not obligate the government to pay coal interests the difference between the government-fixed prices and prices that would otherwise have been obtained.

Skimping on Navy Repairs

The House bill for naval oppropriations for 1923 provides something less than $1 per horsepower per year for supply, repair, maintenance, improvement, inspection, preservation and renewal for the main and auxiliary machinery and boilers, which includes electrical machinery and machine tools, for vessels and shore stations of the Navy. This is less than one-third the average annual allowance per horsepower during the five years immediately preceding the war. The appropriation is approximately $4,500,000 less than the amount requested by the Department, which was intended to provide only for current supplies, operation, and repair without any expenditures for improvements.

The Navy bill is now awaiting passage in the Senate. As it passed the House it does not provide for any work that is on hand to be done in order that the machinery of the fleet may be ready for service. Fully $16,000,000 is needed to bring this machinery up to the standard necessary for immediate duty.

If adequate funds are made available, it will be possible to continue to hold the cost of the maintenance of naval machinery below that of commercial and mercantile ships. This is in spite of the fact that the large power and complexity of naval auxiliary machinery makes for higher costs. The Navy bases its accomplishments in keeping maintenance costs low on the high quality of its machinery and the materials used, and especially on close supervision.

Personals

A. J. Muth, of Chicago, has been appointed to take the place of Joseph H. Stewart, of Goshen, Ind., who resigned June 1 as superintendent of the municipal light, water and heat plant.

Henry C. Buxton, for many years chief engineer of the Singer Building, New York City, became chief engineer of the American Express Building in New York City on June 1, succeeding George R. Warner.

Clarence Griffith has been appointed superintendent of the municipal light and water plant at Muncie, Ind., to take the place of W. E. Murchie, who has resigned to become manager of the electric light plants at Goshen and Warsaw, Ind.

Vreeland Y. Leonard, of Indianapolis, has been made manager of the utilities department of the Interstate Public Service Co., at Columbus, Ind., where he will have charge of the local electric lighting plant. He was formerly consulting engineer and succeeds Kenneth Swan.

John H. Mullen, well known in N.A.S.E. circles, especially in New England, where for years he has arranged the details of the transportation of the delegates to the national conventions, is sojourning in Los Angeles, Cal., where he may be addressed at 1233 Magnolia Ave.

Dr. Howard Monroe Raymond has been appointed President of Armour Institue of Technology. Dr. Raymond is a graduate of the University of Michigan and has been with Armour Institute for the last twenty-seven years. He has been Dean of Engineering since 1903, and has been acting president since the death of Dr. Frank W. Gunsaulus last year.

Charles C. Gardner, of the International Shoe Co., has been transferred from Keene, N. H., where he has started up a new factory, to Morganton, N. C., where he will reorganize the power plants of the sole leather and extract plants there located. He has served the N.A.S.E. as state deputy of Massachusetts and as chairman of the educational committee for the New England States.

Society Affairs

Coming Conventions

American Boiler Manufacturers' Association; H. N. Covell, 191 Dikeman St., Brooklyn, N. Y. Annual convention at Buckwood Inn, Shawnee-on-Delaware, Pa., June 5-7.

American Order of Steam Engineers; Andrew Lauterbach, 4726 North Warnock St., Philadelphia, Pa. Annual convention and exhibition at Philadelphia, June 5-9.

Stoker Manufacturers' Association; J. G. Worker, Eau Claire, Wis. Semiannual meeting at Groton Pt., Conn., June 19-21.

National District Heating Association; D. L. Gaskill, Greenville, Ohio. Annual convention at Cedar Point, Ohio, June 20-23.

American Institute of Electrical Engineers, 29 West 39th St., New York City. Annual convention at Niagara Falls, Ont., June 26-30.

American Society for Testing Materials, 1315 Spruce St., Philadelphia, Pa. Annual meeting at Atlantic City, N. J., June 26-July 1.

Canadian Association of Stationary Engineers; G. C. Keith, 51 Wellington St. West, Toronto, Ont. Annual convention and exhibition at Kitchener, Ontario, June 27-29.

National Association of Stationary Engineers; Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual conventions and exhibitions of the State Associations scheduled as follows:

New Jersey, at Elizabeth, June 2-4; E. J. Flynn, 423 Newpoint Road, Elizabeth, N. J.

Illinois, at Chicago, June 7-10; I. S. Pieters, 1130 Clay St., Chicago.

New York, at New York City, June 8-10; T. J. Condon, 102 Seventh Ave., New York City.

Indiana, at Anderson, June 9-10; Frederick L. Ray, Anderson, Ind.

New England, at Springfield, Mass., July 12-14; James Morgan, 53 Devonshire St., Boston.

Michigan, at Kalamazoo, July 19-21; A. M. Adams, 1307 Stockbridge Ave., Kalamazoo, Mich.

The American Society of Heating and Ventilating Engineers has announced several additions to the program for its semi-annual meeting at Buffalo, June 6-7 and Detroit, June 8-10. The main features of the program were given in *Power* for May 30. An important addition is a paper by F. C. Houghton and L. Ebin on "Capacity of Steam Heating Mains as Affected by Critical Velocities of Condensate Mixtures," to be given June 9.

The American Association for the Advancement of Science will hold a summer session with the sixth annual meeting of the Pacific division and affiliated societies at Salt Lake City, Utah, June 22-24. Arrangements are in charge of the Pacific division, of which the secretary is W. W. Sargent, Golden Gate Park, San Francisco, Cal.

Trade Catalogs

Water Tube Boilers—Henry Vogt Machine Co., Louisville, Ky. Bulletin WT-2, describing and illustrating the company's line of water-tube boilers.

Coal Grinding Mill—Hardinge Co., 120 Broadway, New York City. Condensed bulletin on the Hardinge conical mill, in which coal or other material is ground up by steel balls of different sizes.

Centrifugal Pump Units—Allis Chalmers Manufacturing Co., Milwaukee, Wis. Short bulletin, No. 2061, on the new 1½-in. type "SS" Unit, in which compactness and simplicity are obtained by overhanging the bronze impeller on the motor shaft.

Rotary Pumps—Taber Pump Co., Buffalo, N. Y. Bulletin No. 131, describing belt-, motor-, and engine-driven rotary pumps, giving suggestions for their installation and operation, and containing a useful blueprint chart showing the velocity in feet per second, the friction loss in pounds per square inch, and the friction head in feet, for various quantities of water in gallons per minute passing through 100 ft. of ¾-to 20-in. pipe.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run bases, f.o.b. mines):

Coal	Market Quoting	May 22, 1922	May 29, 1922
Pool 1,	New York	....	....
Smokeless,	Columbus	$3.30	$3.25@3.50
Clearfield,	Boston	3.75	$3.00@3.25
Somerset,	Boston	3.65	3.25@3.50
Kanawha,	Columbus	3.25	3.15@3.35
Hocking,	Columbus	3.65	3.50@3.60
Pittsburgh No. 8	Cleveland	3.90	3.00@3.50
Franklin, Ill.,	Chicago	....	....
Central, Ill.,	Chicago	....	....
Ind. 4th Vein,	Chicago	....	....
West Ky.,	Louisville	3.25	3.10@3.35
Big Seam,	Birmingham	1.70	....
S. E. Ky.,	Louisville	3.40	3.00@3.25

FUEL OIL

New York—June 1, Port Arthur light oil 22@25 deg. Baumé 3⅞c. per gal.; 30@35 deg., 4¾c. per gal., f.o.b. Bayonne, N. J.

Chicago—May 20, for 24@28 deg. Baumé, 85@90c. per bbl.; 21@36 deg., 2⅜@2½c. per gal. in tank cars, f.o.b. Oklahoma refinery, or freight adjusted to conform.

Pittsburgh—May 31, f.o.b. refinery. Pennsylvania, 36@40 deg. 6c. Kentucky fuel oil, 26@30 deg., 4c. per gal. Gas. oil, 32@34 deg., 2½c. per gal.

St. Louis—Mar. 25, prices f.o.b. cars, tank lots; 24@26 deg. Baumé, 90c. per bbl.; 26@28 deg., 95c.; 28@30 deg., $1.10; 32@34 deg. 2¾c. per gal.

Philadelphia—May 29, 26@28 deg. Baumé, Oklahoma, 75@80c. per bbl.; 30@34 deg., Oklahoma (group 3), 2¾@3c. per gal.; 16@20 deg. Seaboard, $1.15@$1.25 per bbl.

Cincinnati—May 27, for 26@30 deg. Baumé, 5½c.; Diesel 32@34 deg., 5½c. per gal.; distillate 38@40 deg., 6c.

Cleveland — May 27, 26@30 deg. Baumé, 4½c. per gal.

New Plant Construction

PROPOSED WORK

Fla., Homestead—The city plans election to vote $50,000 bonds for addition to the electric light plant. B. M. Duncan, Engr.

Ga., Newman—F. B. Cole is in the market for one 3 phase, 60 cycle, 550 volts, 300 to 400 kw. engine and dynamo.

Ill., Bloomington—The Treasury Dept., Supervising Archt. Office, will receive bids until June 19 for installing a steam heating boiler at the U. S. Post Office here.

Ill., Chicago—N. M. Dunning, Archt., 310 South Wabash Ave., is preparing plans for an 8 story, 150 x 171 ft. apartment, including a steam heating system on South Shore Drive. Estimated cost $450,000. Owner's name withheld.

Ill., Chicago—W. Perlman, c/o D. S. Klafter, Archt., 64 West Randolph St., will soon receive bids for a 3 story, 150 x 250 ft. apartment house on Pratt Blvd., between Ashland and Clark Sts. A steam heating system will be installed. About $400,000.

Ill., Evanston—The Main Street Bldg. Corp., c/o J. A. Nyden, Archt., 190 North State St., Chicago, plans to build a 5 story 180 x 187 ft. theater and hotel, including a steam heating system, on Main St. near Chicago Ave. Estimated cost $2,000,000.

Kan., Liberal—The city, L. L. Holland, Clk., is having plans prepared for a power plant, 1,000 hp. capacity. Ruckel Eng. Co., Hutchinson, Engrs.

Mass., Boston—The Statler Hotels Co., E. M. Statler, Pres., Hotel Pennsylvania, 33d St. and 7th Ave., New York, plans to build a hotel on Arlington St., Columbus and St. James Aves. Estimated cost about $8,000,000. G. B. Post & Sons, 101 Park Ave., New York, Engrs.

Mich., Redford—The village plans the construction of a pump house, installation of 1 horizontal motor driven centrifugal pump, capacity 500 GPM, 1 horizontal motor driven centrifugal pumping unit with auxiliary gasoline engine drive, capacity 500 GPM, with auxiliary equipment, also 65,810 ft. 4 to 24 in. watermains. Detroit Water Bd., Detroit, Engrs.

Mo., Kansas City—J. Morse, 1715 Brooklyn St., is in the market for a 30 horsepower, horizontal, high pressure boiler.

Mo., St. Louis—The Masonic Temple Assoc., Railway Exchange Bldg., is having plans prepared for a 5 story, 160 x 300 ft. temple on Lindell Blvd. Estimated cost $2,500,000. A. B. Groves, Stock Exch. Bldg., T. Young, Arcade Bldg., Archts. and E. F. Noble, Fullerton Bldg. Assd. Archt.

Mont., Great Falls—The Toole County irrigation district, J. A. Johnson, Comr., First State Bank, Shelby, plans to build 2 earth dams, four reservoirs, 34,000 ft. of 10-ft. pipe and 7,900 ft. tunnel. Estimated cost $10,000,000. Gerharz & Jaqueth, 514 First Natl. Bank Bldg., Engrs.

Neb., Lincoln—Sanitary District No. 1, Lancaster Co., W. C. Isreal, Secy., will receive bids until June 17 for the construction of sewage disposal plant, including electrically driven centrifugal pumps. Grant, Fulton & Letton, Bankers Life Bldg., and Alvord, Burdick & Howson, 8 South Dearborn St. Chicago, Engrs.

N. J., Bridgeton—Dreher, Churchman & Paul, Archts., Otis Bldg., Phila., are receiving bids for a 5 story, 44 x 117 ft. hotel, with a 46 x 73 ft. wing, on Commerce and Pearl Sts., for the Bridgeton Hotel Co.

N. J., Trenton—The City Comm., will receive bids until June 16 for the installation of 1 Ames unaflow engine, 1 Root rotary pump, electric generator, condensor and atmospheric check valve at pumping station here. Estimated cost $75,000. A. Bugbee, City Hall, Engr. Noted Apr. 4.

N. Y., Albany—The Childs Co., 200 Fifth Ave., New York, is having plans prepared for 6 story restaurant and office building. Dennison & Herons, 475 Fifth Ave., New York, Engrs.

N. Y., Binghamton—O. S. Hathaway, Stone Opera House, plans the construction of a motion picture theater on Washington St. Estimated cost $300,000.

N. Y., Buffalo—The Buffalo General Electric Co., 960 Front St., will soon receive bids for the construction of a 120 x 120 ft. service building on Front St. Estimated cost $135,000. Architect not announced.

N. Y., Buffalo—The Comr. Parks and Buildings, room 6, Municipal Bldg., will receive bids until June 12 for refrigeration machines.

N. Y., Buffalo—The National Biscuit Co., Inc., 217 Ellicott St., is in the market for an electric power plant, air washing, purifying machinery, heating and cooling devices, etc., for new bakery under construction.

N. Y., Buffalo—The E. M. Statler Co. plans to build a 6 story garage on Delaware Ave. and Mohawk St. Estimated cost between $300,000 and $350,000.

N. Y., Lockport—Reder Bros. are in the market for a 25 hp. or larger boiler (new or used).

N. Y., New York—J. Axelrod & Son, representing a syndicate, are having sketches made by G. F. Pelham, Archt. and Engr., 200 West 72nd St., for a 15 story, 90 x 100 ft. apartment hotel at 114-122 East 52nd St. Estimated cost $1,500,000.

N. Y., New York—The George Dose Engineering Co., 565 5th Ave., will build an 18 story hotel on Broadway and West 71st St. Estimated cost $3,000,000. Maynicke & Franke, 25 East 26th St., Archts. and Engrs. Owner will build.

N. Y., New York—The Lapidus Eng. Corp., 17 West 42nd St., will build an 11 story apartment at 23 West 64th St. Estimated cost $500,000. Owner will build.

N. Y., New York—Levoy Bros., c/o Sommerfeld & Steckler, Archts., 31 Union Sq. plans to build offices and store building at 1370 Bway. Estimated cost $700,000. Work will be done by separate contracts under supervision of architect.

N. Y., Niagara Falls—The Bd. Educ. plans construction of a high school. Estimated cost $400,000. W. B. Ittner, 911 Locust St., St. Louis, Mo., and Associated Architects, Buffalo, Archts.

N. Y., Port Henry—The Witherbee Sherman Co. have had plans prepared for additions to plant, including blast furnace, sintering plant, pumping system 9,000,000 gal. daily capacity two 6,500 hp. boilers and 2 blowing engines 45,000 cu.ft. each.

N. Y., Port Jervis—The Bd. Educ. is having plans prepared for a high school. Estimated cost $400,000. Tooker & Marsh, 101 Park Ave., New York, Engrs.

N. Y., Rochester—The Bd. Educ. plans to build a 166 x 186 ft. addition to School No. 43 on Field Road. Estimated cost $475,000. Heating and ventilating system will be separate contract. Architect not announced.

N. C., Greensboro—P. C. Painter, City Mgr., will receive bids until June 8 for building dam across Reedy Fork Creek to develop an impounding reservoir and for furnishing and installing an auxiliary power plant. J. L. Ludlow, Winston Salem, Engr.

N. C., Hendersonville—The Bd. Water Comrs., C. E. Brooks, Clk., will receive bids until June 15 for dam and reservoir and 15 mi. in water pipe. G. C. White, Durham, Engr.

N. C., Oteen—The Treas. Dept., J. A. Wetmore, Acting Supervising Archt., Wash., D. C., will receive bids until June 16 for installing 2 horizontal return tubular, high pressure, steam boilers with hand stokers at the U. S. Veteran Hospital here.

Ohio, Chilo—The U. S. Engineers Office, 405 Custom House, will receive bids until June 19 for the construction of Dam No. 24 on the Ohio River here.

Ohio, Cleveland—Catholic Diocese of Cleveland c/o Rev. J. Schremba, Bishop, East 9th St., and Superior Ave., plans to build 25 churches and schools in various sections. Estimated cost $10,000,000. Architect not selected.

Ohio, Cleveland—The Cleveland Aviation & Athletic Club, H. W. Harris, Vice Pres., Winton Hotel, plans to build an office building. Estimated cost $5,000,000. A. W. Harris, Discount Bldg., Archt.

Ohio, Columbus—The Bd. Educ., E. L. McCune, Clk., Towne and High Sts., will receive bids until June 16 for 2 story, 278 x 327 ft. Washington Gladden high school on Arcadia Ave., F. L. Packard, Archt., New Hayden Bldg., also 3 story, 200 x 411 ft., E. J. Wald Orton high school on the Scioto river, W. B. Ittner, 911 Locust, Mo., Archt. Estimated cost $870,000 and $1,000,000 respectively.

Ohio, Dayton—J. F. Gibbons, Gibbons Hotel, plans to construct a 20 story, 98 x 165 ft. hotel at 3rd and Main Sts. About $3,000,000. Architect not selected.

Ohio, Salem—The city will soon receive bids for the construction of a reservoir, tank house, pipe lines, etc., for waterworks system. Estimated cost $500,000. M. Knowles, Jones Law Bldg., Pittsburgh, Pa., Engr. Noted Apr. 4.

Pa., McKees Rocks—The Chesebrough Mfg. Co., 17 State St., New York City is having plans prepared for a manufacturing plant, consisting of 7 buildings including power house, boiler house and warehouse. Estimated cost $500,000. Hunting Davis Co., Century Bldg., Pittsburgh, Pa., archts.

Pa., Newton Falls—The Newton Steel Co., is having plans prepared for the construction of 6 sheet mills. Estimated cost $750,000. Want quotation on electrical machinery for operating sheet steel mills. Engineer not announced.

Pa., Phila.—The Bureau of Water, 696 City Hall, will receive bids until June 8 for 6 motor-driven air compressors, 2 stoker engines, 3 steam turbine governors, 1 steam jet air pump, 1 turbine driven pump and 4 motor driven pumps. C. Davis, City Hall, Engr.

Pa., Phila.—The Manchester Steam Laundry is in the market for a 50 kw. d.c. generator directly connected to a horizontal steam engine.

S. D., Huron—The city, c/o W. E. Nelson, Aud., is having plans prepared for increasing the capacity of their light and power plant; installing a street lighting system; adding 1 Diesel engine, 2 generators, etc. Estimated cost $100,000. J. H. A. Brahtz, 409 Metropolitan Opera House Bldg., St. Paul, Engr.

S. D., St. Lawrence—The town clerk will receive bids until June 15 for the construction of a 50,000 gal. reservoir and 150,000 gal. centrifugal pump and motor. Dakota Engineering Co., Mitchell, Engrs.

Texas, San Antonio—San Antonio Amusement Co., c/o W. J. Lytle, Princess Theater, are having plans prepared for 2 story 125 x 150 ft. theater building, including heating and power plant on Houston and Travis Sts. Estimated cost $1,000,000. Prince & Benjamin, Dallas, Archts.

Tex., Waco—The Liberty Natl. Bank plans to build an 8 story bank and office building, Austin and Capital Sts., B. D. Esterwood, 412 Pretorian Bldg., Archt.

Va., Portsmouth—City and Norfolk County, J. P. Jervey, City Mgr., plans to build a city and county office building on Court St. including courthouse and auditorium. Estimated cost $800,000. Architect not selected.

Va., West Point—The Chesapeake Corp. of West Point, W. C. Gouldman, Secy., plans construction of a mill to manufacture pulp board and paper specialties. Estimated cost $200,000 to $250,000.

Wash., Monroe—The city voted $84,000 bonds to build a waterworks system, including a 1,000,000 gal. reservoir. Estimated cost $80,000. The Miller Eng. Co., Burke Bldg., Seattle, Engrs.

W. Va., Bluefield—The American Fruit Co., H. Charlton, Vice Pres., plans to build a 6 story addition to cold storage plant. Estimated cost, including machinery $250,000. Quotations wanted on machinery.

W. Va., Rivesville—The town plans to install a waterworks system, 40,000 gal. capacity including centrifugal pumps, motors, etc., also a 50,000 gal. reservoir. Estimated cost $40,000. B. Kress, Elkins, Engr.

Wis., Manitowoc—The Bd. Educ., E. Kerwan, Secy., will receive bids until June 15 for a 3 story, 240 x 419 ft. high school, including a steam heating system. Estimated cost $400,000. Perkins, Fellows & Hamilton, 814 Tower St., Chicago, Archts. Noted March 28.

Wis., Milwaukee—The Sewerage Comn., J. H. Fowles, Secy., 580 Market St., will receive bids until June 16 for 90 x 225 ft. and 68 x 166 ft., superstructures of main power house and boiler plant on Jones Island T. C. Hatton, c/o Sewage Comn., Engr.

Wis., Wauwatosa—The Bd. Trustees of Milwaukee County is having plans prepared by F. A. Vaughn, Inc., Engrs., 290 3rd St., Milwaukee, for a heating and power plant to supply all county owned institutions W. L. Coffey, Mgr.

Ont., Belle River—The Council, A. La Charite, is having plans prepared for an electric lighting and power distribution system. Estimated cost $35,000. E. D. Lawler, 190 University Ave., Toronto, Engr.

Ont. London—The Utilities Comrs., c/o E. V. Buchanan, Engr., Hydro Offices, is in the market for two 1,500 kw. transformers.

Ont., Ottawa—The Ottawa Hydro Electric Comn., J. E. Brown Gen. Mgr., 109 Bank St., will soon award the contract for structural steel for substation on Carling Ave. All electrical units to be purchased by Ottawa Hydro Electric Comn. Total estimate $180,000.

Ont., Thessalon—The ratepayers have voted in favor of a bylaw for $50,000 to develop 200 hp. at Little Rapids on the Thessalon River. Project will include dam, power house generator, transformers, switches, transmission line, etc. J. O. Coulter, Clk.

T. H., Pearl Harbor—Bureau of Yards & Docks, Navy Dept., plans extension to power houses here and at Heeia.

CONTRACTS AWARDED

Conn., New Britain—The Bd. of Water Comrs. has awarded the contract for the construction of a caisson well, pumping station, etc., to the Eastern Engineering & Constr. Co., 945 Main St., Bridgeport, $49,454.

Ill., Chicago—The Lake Shore Drive Hotel Bldg. Corp., c/o Fugard & Knapp, Archts., 64 East Van Buren St., has awarded the contract for a 19 story, 120 x 210 ft. hotel, on Lake Shore Dr. and Walton Pl. to the McLennon Constr. Co., 400 North Michigan St. Estimated cost $3,000,000. A steam heating system will be installed.

Mass., Boston—The Traders Building Trust, 18 Tremont St., has awarded the contract for the construction of an 11 story mercantile building on Kneeland St., to Snider, Druker & Rodnick, 18 Tremont St. Estimated cost $500,000. Steam heating and ventilating system will be installed. Noted May 30.

N. Y., Bay Shore, L. I.—The South Side Hospital has awarded the contract for a 4 story 100 x 100 ft. hospital building to W. L. Crow Constr. Co., 103 Park Ave., New York. Estimated cost $300,000.

N. Y., New York—New York Cotton Exchange, 107-111 Hanover Sq., has awarded the contract for a 23 story 90 x 115 ft. bank and office building on Hanover Sq. to G. A. Fuller Co., 949 Bway. Estimated cost $2,250,000. Noted Aug. 2, 1921.

N. Y., New York—The N. Y. Edison Co., Irving Pl. and 15th St., has awarded the contract for a sub station on 40th St. and 1st Ave., to the W. L. Crow Constr. Co., 103 Park Ave. Estimated cost $250,000.

Ohio, Cleveland—The Union Trust Co., c/o M. S. Halliday, 1128 Citizens Bldg., has awarded the contract for a 21 story, 147 x 258 ft. bank and office building on East 9th St. and Euclid Ave., to Thompson-Starrett Co., 175 Jackson Blvd., Chicago. About $10,000,000. Noted Jan. 31.

Pa., Danville—The State has awarded the contract for 150 x 300 ft. power plant to develop 300 hp. at state hospital here, to McClintic Marshall Co., Oliver Bldg., Pittsburgh.

Pa. Newton Square—C. Brazer, Archt., Chester, has awarded the contract for institution buildings, power house, etc., for the Dunwoody Home to Barclay & White Co., 1713 Sansom St., Phila. About $300,000. Noted Apr. 4.

Pa., Phila.—Roop & Baschore, c/o Weber & Wurster, Archts., Morris Bldg., has awarded the contract for 15 story, 69 x 163 ft. office building, including an indirect heating system, on 1015 Chestnut St., to P. H. Kelly Co., 1713 Sansom St. Estimated cost $2,500,000.

Wis., Milwaukee—The Milwaukee Elec. Ry. & Lt. Co., Public Service Bldg., has awarded the contract for 1 story, 60 x 40 and 50 x 60 ft. additions to turbine room and switchhouse at Lakeside, to Meredith Bros., 1043 Kinnicinnic St.

Wis., Monroe—The Badger Cheese Co. has awarded the contract for a 2 story 26 x 100 ft. cold storage plant to L. S. Wagner & Son. Estimated cost $25,000. Refrigeration equipment will be installed.

Ont., Toronto—The Conger Coal Co., Ltd., 95 Bay St., has awarded the contract for a coal handling plant on Main St. to J. B. Nicholson & Son, 36 Toronto St. About $20,000.